POWER

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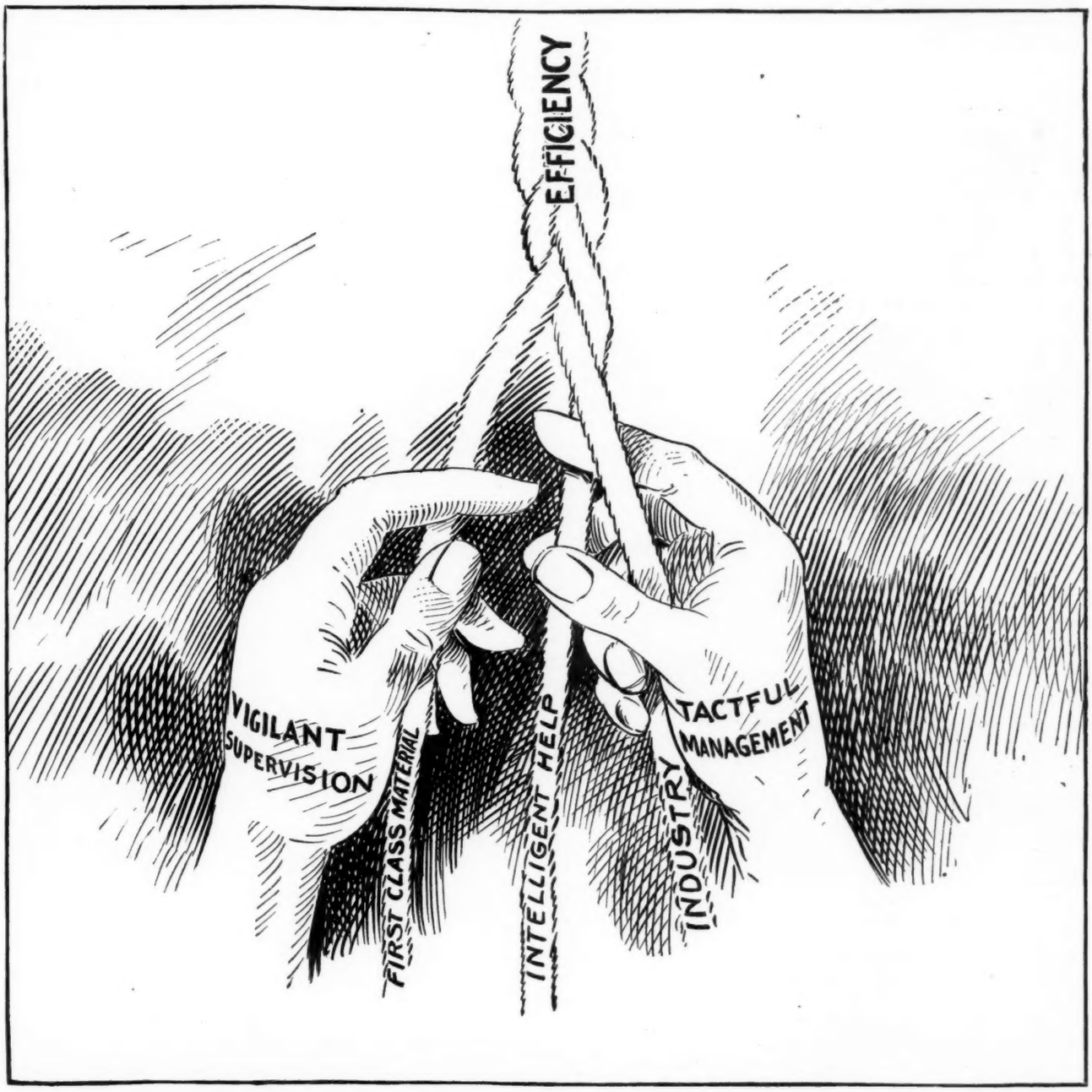

CAUSE AND EFFECT

Flat Surfaces Supported by Stay Bolts

By A. J. Toppin

A set of tables and curves showing the safe working pressures on flat stayed surfaces with different pitches and diameters of stay-bolts. They are computed for both the Massachusetts and the United States Government rules.

To save laborious calculations on the part of the boiler inspector and the boiler maker the writer has computed a set of tables giving the safe working pressure on the flat stayed surfaces of a boiler. By a flat surface is meant that which has a curvature above a certain radius; in Massachusetts, Ohio and the city of Detroit this radius is 19 in. and the United States Government specifies it as 21 in. A true flat surface is one which has an infinite radius.

The staying of flat surfaces resolves into two separate problems: the allowable pressure for a given thickness of plate and pitch, and the size of stay-bolt necessary to support the load. For illustration, consider the parallel case of a truss bridge spanning a river. No matter how strong the piers may be, if the members are not strong enough for the span and load, the structure will collapse, the piers remaining intact. On the other hand, although the structure may be strong enough for the given span and load, if the piers are not properly built, the truss will drop into the river.

Reverting to flat surfaces, the methods of failure are illustrated in Figs. 1, 2 and 3. A plate which is too thin would bulge as in Fig. 1; whereas a stay-bolt can fail in either of two ways, by breaking across its weakest section—the root of the threads (Fig. 2)—or by stripping its threads and riveted head (Fig. 3).

First Case

There seems to be a difference of opinion relative to the allowable pressure for a given thickness of plate and pitch of rivets. There are formulas by different authorities which give notably different results, as will be pointed out later when discussing the curves plotted. Massachusetts, Ohio and the city of Detroit employ the formula:

$$P = \frac{66\,(t+1)^2}{S^2 - 6}$$

in which,

P = Safe working pressure in pounds per square inch;
t = Thickness of plate in sixteenths of an inch;
66 = A constant determined by experiment;
S = Maximum pitch in inches.

The United States Government rule is:

$$P = \frac{k \times t^2}{S^2}$$

in which P, t and S are the same symbols as in the previous formula and k is a constant depending on the method of staying. For screwed stays riveted over and plates not exceeding $\frac{7}{16}$ in. thick, k = 112. For the same conditions and plates over $\frac{7}{16}$ in. thick, k = 120. There is also a formula by D. K. Clark:

$$P = \frac{k \times t}{l}$$

in which

P = Pressure in pounds per square inch, which will strain the plate to its elastic limit.
k = A constant, 5000 for iron and 5700 for steel;
t = Thickness of plate in inches;
l = Maximum distance in the clear.

By "distance in the clear" is meant from

Fig. 1. Failure by Bulging

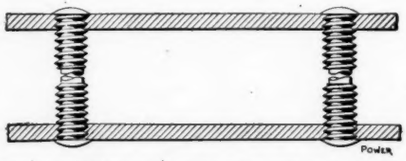

Fig. 2. Breaking of Stay-bolt at Root of Thread

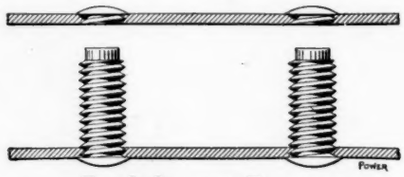

Fig. 3. Stripped Threads

rot of thread to root of thread. If d is the diameter at the root of the thread, then l = Pitch — d (see Fig. 5).

The latter formula does not appear to be employed as universally as the other two but curves are plotted from it for comparison. It will be noted that this does not give the safe working pressure direct, but must be divided by a factor. As the elastic limit is generally about one-half the breaking load, the factor should be one-half the factor of safety based on the ultimate tensile strength. In other words, if a factor of safety of 6 is ordinarily employed the pressure found by the formula should be divided by 3 to get the safe working pressure. This factor was used in plotting the curves. In Clark's formula the factor of safety is chosen, while with the other two it is already contained within them and cannot be varied. Also in Clark's formula distinction to a small extent is made as regards the material, as he allows a constant of 5000 for iron and 5700 for steel plates.

Most authorities consider flat surfaces the same as beams fixed at each end and uniformly loaded, and claim that the formula for flat surfaces should follow the general formula for flexure of beams, constants being determined by experiment. Others claim that the formula should not follow these lines inasmuch as the fibers are all in tension and not half in tension and half in compression as in beams. In the writer's opinion the Massachusetts and United States Government formula and several others are based on the former reasoning and Clark's formula on the latter.

Second Case

The second problem is much simpler than the first: To determine the area of bolt necessary to sustain the load, let

P = Pressure in pounds per square inch;
$S_{max.}$ = Maximum pitch in inches;
$S_{min.}$ = Minimum pitch in inches;
A = Net area under pressure, in square inches;
D = Outside diameter of stay-bolt in inches;
d = Diameter at root of thread of stay-bolt.

The net area supported by one stay-bolt would be the product of the two pitches minus the area of stay-bolt at the root of the thread, which expressed in symbols would be

$$A = (S_{max.} \times S_{min.}) - \left(\frac{\pi d^2}{4}\right)$$

The load sustained by the stay-bolt would be this area times the pressure or

$$P \times \left[(S_{max.} \times S_{min.}) - \left(\frac{\pi d^2}{4}\right)\right]$$

Let T = the allowable tensile strength of the stay-bolt in pounds per square inch. Then the strength of the stay-bolt would be $\left(T \times \frac{\pi d^2}{4}\right)$. As this must balance the load

$$T \times \frac{\pi d^2}{4} = P\left[(S_{max.} \times S_{min.}) - \left(\frac{\pi d^2}{4}\right)\right]$$

Transposing,

$$P = \frac{\frac{T \pi d^2}{4}}{S_{max.} \times S_{min.} - \frac{\pi d^2}{4}} = \frac{T \pi d^2}{4\,S_{max.} \times S_{min.} - \pi d^2}$$

or

$$d = 2\sqrt{\frac{P S_{max.} \times S_{min.}}{\pi\,(T+P)}}$$

Stay-bolts 1¼ in. in diameter and under, generally have 12 threads per inch and the outside diameter

$$D = d + 0.1443$$

or

$$d = D - 0.1443$$

Substituting in the foregoing

$$D - 0.1443 = 2\sqrt{\frac{P S_{max.} \times S_{min.}}{\pi (T + P)}}$$

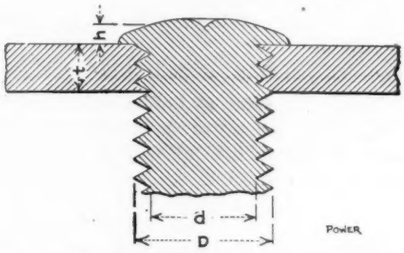

FIG. 4. REPRESENTING SHEARING RESISTANCE OF HEAD

or

$$D = 2\sqrt{\frac{P S_{max.} \times S_{min.}}{\pi (T + P)}} + 0.1443$$

The value of T varies according to different authorities. Massachusetts allows only 6500 lb. per square inch on mild steel or wrought-iron stay-bolts up to and including 1¼ in. diameter, whereas the United States Government allows 6000.

Failure by stripping the threads is remote and is therefore not calculated but a formula can be deduced for determining the height of the head of the stay-bolt. The threads would strip if the plate into which the bolt is screwed was too thin to afford the proper number of threads to have a hold, unless it is sufficiently headed over to make up the deficiency. The shearing surface furnished by the head would be $3.1416 \times d \times h$, as shown in Fig. 4. The area to resist shearing the

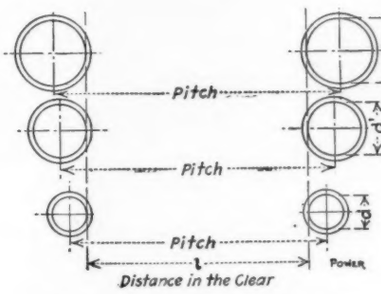

FIG. 5. DISTANCE IN THE CLEAR

threads would be $3.1416 \times d \times t$. The total area to resist stripping would be:

$$(3.1416 \times d \times h) + (3.1416 \times d \times t) = 3.1416\, d\, (h + t)$$

Multiplying this by the shearing value per square inch of the metal in the bolt would give the stress necessary to strip the threads. Calling this value *S.S.*, the stress would be $3.1416\, d\, (h + t)\, S.S.$ This must be at least equal to the stress necessary to break the bolt at the root of the thread, which would be $\left(\frac{3.1416\, d^2}{4}\right) \times T.S.$ in which *T.S.* is the ultimate tensile strength per square inch of the metal in the bolt. Then

$$\frac{3.1416\, d^2}{4} \times T.S. = 3.1416\, d\, (h + t)\, S.S.$$

or

$$\frac{d \times T.S.}{4} = (h + t)\, S.S. = h \times S.S. + t \times S.S.$$

and

$$h = \frac{d\, T.S.}{4\, S.S.} - t$$

The value of *S.S.* varies from 35,000 to 55,000 lb. per square inch for wrought iron and from 45,000 to 75,000 for steel. This formula shows that with constant thickness of plate, the height of head should be increased as the diameter of the bolt is increased. Keeping the diameter constant the height of the head should be increased as the thickness of plate is decreased. If the shearing value of the metal of the plate was less than that of the bolt and there was no head to the bolt the threads of the plate would strip.

STAY-BOLTS WITH TELLTALE HOLES

The United States rules call for a ⅛-in. telltale hole drilled in the ends of each stay-bolt. This, of course, reduces its strength in direct proportion to the area of the bolt. The area of a ⅛-in. circle is 0.01227 sq.in., and the area of a ¾-in. bolt at the root of the thread is 0.288 sq.in. The percentage of reduction of area would be

$$\frac{0.01227}{0.288} \times 100 = 4.26 \text{ per cent.}$$

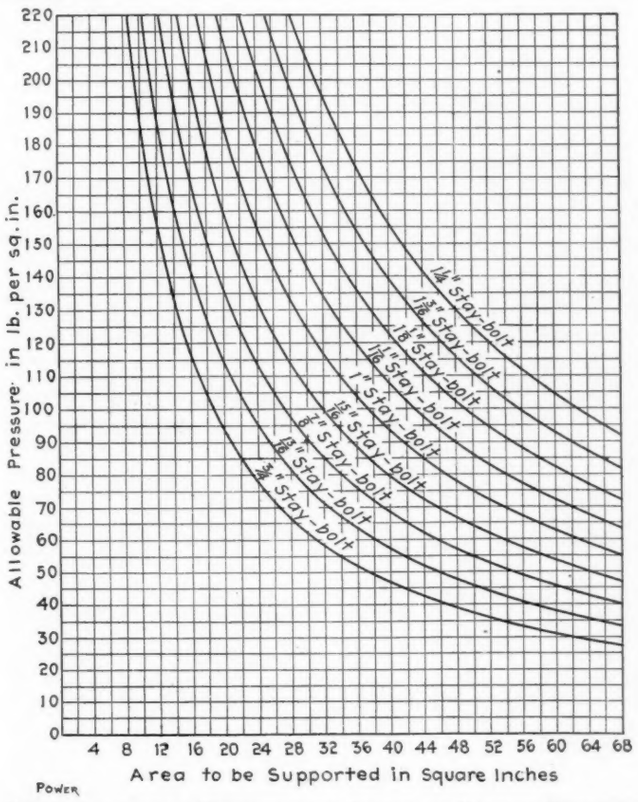

FIG. 6. ALLOWABLE PRESSURES ON PLATES OF VARIOUS THICKNESS AND PITCHES—FACTOR OF SAFETY OF 6 USED

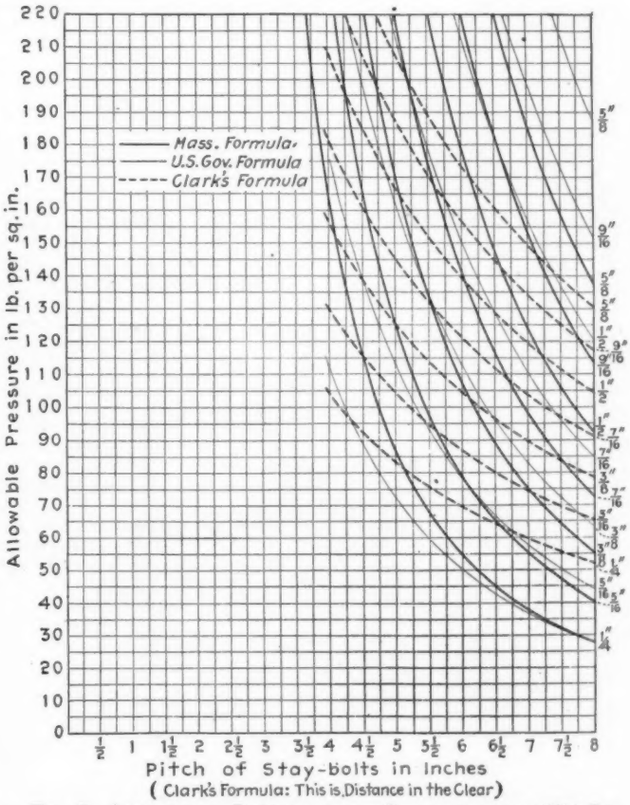
Mass. Formula
U.S. Gov. Formula
Clark's Formula
Allowable Pressure in lb. per sq. in.
Pitch of Stay-bolts in Inches
(Clark's Formula: This is Distance in the Clear)

FIG. 7. ALLOWABLE PRESSURES ON STAY-BOLTS AT 6500 LB. STRESS

Thickness of Plate in Inches	Allowable Pressures on Stay-bolted Flat Plate																							Varying Pitch and Thickness									
	Maximum Pitch in Inches																																
	4	4⅛	4¼	4⅜	4½	4⅝	4¾	4⅞	5	5⅛	5¼	5⅜	5½	5⅝	5¾	5⅞	6	6⅛	6¼	6⅜	6½	6⅝	6¾	6⅞	7	7⅛	7¼	7⅜	7½	7⅝	7¾	7⅞	8
¼	165	149	136	125	115	107	99	93	86	81	76	72	68	64	61	57	55	52	49	47	45	43	41	40	38	36	35	34	32	31	30	29	28
5/16	237	215	197	180	166	154	143	133	125	117	110	103	98	92	87	83	79	75	71	68	65	62	60	57	55	53	51	49	47	45	43	42	40
⅜					227	210	195	182	170	160	150	141	133	126	119	113	107	102	97	93	89	85	81	78	75	72	69	66	64	62	59	57	55
7/16									222	208	195	184	174	164	156	148	140	134	127	121	116	111	106	102	98	94	90	87	84	81	78	75	72
½													220	208	197	187	178	169	161	154	147	141	135	129	124	119	114	110	106	102	98	95	92
9/16																	220	209	199	190	182	174	166	159	153	147	141	136	131	126	122	117	113
⅝																					220	210	201	193	185	178	171	164	158	153	147	142	137

The Above Table was Calculated According to Massachusetts Formula

$$P = \frac{66\,(t+1)^2}{S^2-6}$$

In Which P = Allowable Pressure in lb. per sq.in.
S = Maximum Pitch in Inches
t = Thickness of Plate in Sixteenths of an Inch
66 = A Constant determined by Experiment

TABLE. 1. ALLOWABLE PRESSURES ON STAY-BOLTED FLAT SURFACES ACCORDING TO MASSACHUSETTS FORMULA

Thickness of Plate in Inches	Allowable Pressures on Stay-bolted Flat Plates																							Varying Pitch and Thickness									
	Maximum Pitch in Inches																																
	4	4⅛	4¼	4⅜	4½	4⅝	4¾	4⅞	5	5⅛	5¼	5⅜	5½	5⅝	5¾	5⅞	6	6⅛	6¼	6⅜	6½	6⅝	6¾	6⅞	7	7⅛	7¼	7⅜	7½	7⅝	7¾	7⅞	8
¼	112	105	99	93	88	83	79	75	71	68	65	62	59	56	54	51	49	47	45	44	42	40	39	37	36	35	34	32	31	30	29	28	28
5/16	175	164	155	146	138	134	124	117	112	106	101	96	92	88	84	81	77	74	71	68	66	63	61	59	57	55	53	51	49	48	46	45	43
⅜			223	210	199	188	178	169	161	153	149	139	133	127	122	116	112	107	103	99	95	91	88	85	82	79	76	74	71	69	67	65	63
7/16								226	215	205	195	186	178	170	163	156	149	143	138	132	127	122	118	114	109	106	102	99	95	92	89	86	84
½																230	213	204	196	188	181	174	168	162	156	151	146	141	136	132	128	123	120
9/16																						221	215	205	198	191	184	178	172	167	161	156	151
⅝																												220	213	216	199	193	187

The Above Table Calculated According to the U.S.Gov. Formula $P = \frac{kt^2}{S^2}$ In Which k = A Constant, S = Max. Pitch, t = Thickness of Plate

k = 112 For Plates up to $\frac{7}{16}''$, k = 120 For Plates over $\frac{7}{16}''$

TABLE 2. ALLOWABLE PRESSURES ON STAY-BOLTED FLAT SURFACES ACCORDING TO U. S. GOVERNMENT FORMULA

or the area is 95.74 per cent. of what a solid ¾-in. bolt would be. This may be expressed as a constant 0.957 by which the values given in the tables should be multiplied.

Size of Stay-bolt	Stress in Pounds per Square Inch				
	6000	6500	7000	7500	8000
¾"	0.882	0.957	1.03	1.10	1.17
13/16"	0.888	0.965	1.03	1.11	1.18
⅞"	0.893	0.970	1.04	1.12	1.19
15/16"	0.898	0.975	1.05	1.12	1.20
1"	0.902	0.979	1.05	1.13	1.20
1 1/16"	0.903	0.981	1.05	1.13	1.20
1 ⅛"	0.906	0.983	1.05	1.13	1.21
1 3/16"	0.907	0.985	1.06	1.13	1.21
1 ¼"	0.909	0.987	1.06	1.14	1.21

TABLE 3. CONSTANTS FOR STAY-BOLTS WITH ⅛-IN. TELL-TALE HOLES TO BE USED IN CONJUNCTION WITH STAY-BOLT TABLES

Suppose the allowable pressure on a 1-in. stay-bolt, with the pitch 5x6 in., and having a ⅛-in. telltale hole is desired, the allowable stress being taken at 7000 lb. per square inch.

Table 7, for a 1-in. solid bolt, at 6500 lb. safe tensile stress, gives 127 lb. as the allowable unit stress.

The constant for a 1-in. bolt with ⅛-in. hole is 0.979 (Table 3) and the constant for 7000 lb. stress is 1.077. Therefore the allowable unit pressure would be

$$0.979 \times 1.077 \times 127 = 133\ lb.$$

These two constants may be combined and to save multiplying by two numbers Table 3 has been compiled, giving a single constant combining both for different sized bolts and various stress allowances. The constant, for instance, for the foregoing problem would be

$$0.979 \times 1.077 = 1.05$$

This constant multiplied by the pressure would be

$$127 \times 1.05 = 133$$

which is the same as before. Sometimes the telltale holes are 3/16 in. diameter, so that constants are also given for this size hole; see Table 4.

Size of Stay-bolt	Stress in Pounds per Square Inch				
	6000	6500	7000	7500	8000
¾"	0.833	0.904	0.975	1.045	1.112
13/16"	0.848	0.921	0.992	1.064	1.133
⅞"	0.862	0.935	1.008	1.081	1.151
15/16"	0.871	0.945	1.018	1.092	1.163
1"	0.876	0.952	1.024	1.100	1.172
1 1/16"	0.882	0.957	1.031	1.108	1.179
1 ⅛"	0.888	0.963	1.038	1.113	1.187
1 3/16"	0.893	0.967	1.042	1.119	1.191
1 ¼"	0.893	0.971	1.045	1.122	1.195

TABLE 4. CONSTANTS FOR STAY-BOLTS WITH 3/16-IN. TELL-TALE HOLES TO BE USED IN CONJUNCTION WITH STAY-BOLT TABLES

EXAMPLES

The boiler inspector and the boiler maker approach the stay-bolt problem from opposite points of view. The latter sets out to build a boiler for a given pressure, and desires to determine the pitch and size of stay-bolts for the firebox. The inspector has these quantities before him and desires to determine the permissible pressure.

Suppose a boiler maker is to use a 5/16-in. plate and desires the pitch and size of stay-bolts for 100 lb. pressure according to the Massachusetts rules. Consulting Table 1 it will be found that a pitch of 5⅜ in. is good for 103 lb.

¾" Stay-bolt Allowable Pressures at 6500 lb. per Sq. In. Stress — Area of Bolt at Root of Threads = 0.288 Sq. In. 12 Threads per Inch

Pitch	8	7⅞	7¾	7⅝	7½	7⅜	7¼	7⅛	7	6⅞	6¾	6⅝	6½	6⅜	6¼	6⅛	6	5⅞	5¾	5⅝	5½	5⅜	5¼	5⅛	5	4⅞	4¾	4⅝	4½	4⅜	4¼	4⅛	4
4																									95	97	100	102	105	108	112	115	119
4⅛																								89	92	94	97	99	102	105	108	111	
4¼																							85	87	89	91	94	96	99	102	105		
4⅜																						80	82	84	86	88	91	93	96	99			
4½																					76	78	80	82	84	86	88	91	93				8
4⅝																				72	74	76	78	79	82	84	86	88					7⅞
4¾																			69	70	72	74	75	77	79	81	84						7¾
4⅞																		66	67	69	70	72	73	75	77	79							7⅝
5																	63	64	65	67	68	70	72	73	75								7½
5⅛																60	61	62	64	65	67	68	70	72									7⅜
5¼															57	58	59	61	62	64	65	67	68										7¼
5⅜														55	56	57	58	59	61	62	64	65			45								7⅛
5½													52	53	54	56	57	58	59	61	62			46	46								7
5⅝												50	51	52	53	54	55	57	58	59			48	47	46								6⅞
5¾											48	49	50	51	52	53	54	55	57			50	49	48	47								6¾
5⅞										46	47	48	49	50	51	52	53	54			52	51	50	49	48								6⅝
6									44	45	46	47	48	49	50	51	52			54	53	52	51	50	49								6½
6⅛								43	43	44	45	46	47	48	49	50			56	55	54	53	52	51	50								6⅜
6¼								42	43	43	44	45	46	47	48			58	57	56	55	54	53	52	51								6¼
6⅜								41	42	42	43	44	45	46			61	60	59	57	56	55	54	53	52								6⅛
6½								40	41	42	42	43	44			64	62	61	60	59	57	56	55	54									6
6⅝								39	40	41	42	42			66	65	64	62	61	60	59	58	57										5⅞
6¾								39	39	40	41			69	68	66	65	64	62	61	60	59											5¾
6⅞								38	39	39			72	71	69	68	66	65	64	63	61												5⅝
7								37	38			76	74	72	71	69	68	67	65	64													5½
7⅛								37			79	78	76	74	73	71	70	68	67														5⅜
7¼										83	81	80	78	76	74	73	71	70															5¼
7⅜									88	85	83	81	80	78	76	75	73																5⅛
7½								92	90	88	86	84	82	80	78	76																	5
7⅝							97	95	92	90	88	86	84	82	80																		4⅞
7¾						102	100	97	95	93	90	88	86	84																			4¾
7⅞					108	105	102	100	97	95	93	90	88																				4⅝
8				114	111	108	105	103	100	98	95	93																					4½
			121	117	114	111	109	106	103	100	98																						4⅜
		128	125	121	118	115	112	109	106	103																							4¼
	137	132	128	125	121	118	115	112	109																								4⅛
145	141	137	133	129	125	122	119	116																									4
4	4⅛	4¼	4⅜	4½	4⅝	4¾	4⅞	5	5⅛	5¼	5⅜	5½	5⅝	5¾	5⅞	6	6⅛	6¼	6⅜	6½	6⅝	6¾	6⅞	7	7⅛	7¼	7⅜	7½	7⅝	7¾	7⅞	8	Pitch

13/16" Stay-bolt Allowable Pressures at 6500 lb. per Sq. In. Stress — Area of Bolt at Root of Threads = 0.351 Sq. In. 12 Threads per Inch

TABLE 5. ALLOWABLE PRESSURES ON ¾- AND 13/16-IN. STAY-BOLTS

Pitch	7/8" Stay-bolt Allowable Pressures at 6500 lb. per sq. in. Stress																															Area of Bolt at Root of Threads = 0.419 sq. in. 12 Threads per Inch
	8	7 7/8	7 3/4	7 5/8	7 1/2	7 3/8	7 1/4	7 1/8	7	6 7/8	6 3/4	6 5/8	6 1/2	6 3/8	6 1/4	6 1/8	6	5 7/8	5 3/4	5 5/8	5 1/2	5 3/8	5 1/4	5 1/8	5	4 7/8	4 3/4	4 5/8	4 1/2	4 3/8	4 1/4	4 1/8	4	
4																									139	142	146	150	154	159	164	169	178
4 1/8																									131	134	138	142	146	150	154	159	164
4 1/4																									124	127	130	134	137	141	145	149	154
4 3/8																									118	120	123	127	130	133	137	141	145
4 1/2																									111	114	117	120	123	126	130	133	137
4 3/8																									106	108	111	114	117	119	123	126	129
4 3/4																									101	103	105	108	111	113	116	119	122
4 7/8																									96	98	100	103	105	108	110	113	116
5																									92	94	96	98	100	103	105	108	110
5 1/8																									88	89	91	93	95	98	100	102	105
5 1/4																									84	85	87	89	91	93	95	98	100
5 3/8																										80	82	83	85	87	89	91	93	95
5 1/2																										77	78	80	81	83	85	87	89	91
5 3/4																											73	75	76	78	80	81	83	85	87
5 7/8																											70	72	73	75	76	78	79	81	83
6																												68	69	70	72	73	75	76	78	79
6 1/8																													65	66	68	69	70	72	73	75	76
6 1/4																													63	64	65	66	67	69	70	72	73
6 3/8																													61	62	64	65	66	67	69	70	
6 1/2																													60	61	62	63	65	66	67		
6 3/4																													59	60	61	62	63	65			
6 7/8																													58	59	60	61	62				
7																													57	58	59	60					
7 1/8																													56	57	58						
7 1/4																													55	56							
7 3/8																													54								
7 1/2																																					
7 5/8																																					
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Pitch	15/16" Stay-bolt Allowable Pressures at 6500 lb. per sq. in. Stress																																	Area of Bolt at Root of Threads = 0.494 sq. in. 12 Threads per Inch
	8	7 7/8	7 3/4	7 5/8	7 1/2	7 3/8	7 1/4	7 1/8	7	6 7/8	6 3/4	6 5/8	6 1/2	6 3/8	6 1/4	6 1/8	6	5 7/8	5 3/4	5 5/8	5 1/2	5 3/8	5 1/4	5 1/8	5	4 7/8	4 3/4	4 5/8	4 1/2	4 3/8	4 1/4	4 1/8	4	
4																									192	197	203	208	214	221			
4 1/8																									181	186	191	196	202	207	214	220	
4 1/4																									171	176	180	185	190	195	201	207	213
4 3/8																									162	166	171	175	180	185	190	195	201
4 1/2																									154	158	162	166	170	175	179	184	190
4 5/8																									146	150	153	157	161	165	170	174	179
4 3/4																									139	142	146	149	153	157	161	165	170
4 7/8																									133	136	139	142	145	149	153	157	161
5																									127	129	132	135	139	142	145	149	153
5 1/8																									121	123	126	129	132	135	138	142	145
5 1/4																									116	118	120	123	126	129	132	135	138
5 3/8																									110	113	115	118	120	123	126	129	132
5 1/2																									106	108	110	112	115	117	120	123	125
5 3/8																									101	103	105	108	110	112	115	117	120
5 3/4																									97	99	101	103	105	107	110	112	115
5 7/8																									93	95	97	99	101	103	105	107	110
6																									90	91	93	95	97	99	101	103	105
6 1/8																									86	88	90	91	93	95	97	99	101
6 1/4																									85	86	88	89	91	93	95	97	
6 3/8																									83	84	86	88	89	91	92		
6 1/2																									81	83	84	86	88	89			
6 3/8																									80	81	83	84	86				
6 3/4																									78	80	81	83					
6 7/8																									77	78	80						
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TABLE 6. ALLOWABLE PRESSURES ON 7/8- AND 15/16-IN. STAY-BOLTS

Next consulting the stay-bolt tables it will be found that using equal horizontal and vertical pitches (5 3/8 x 5 3/8 in.) a 15/16-in. stay-bolt would be necessary. This stay-bolt is slightly stronger than necessary, being good for 113 lb. If, however, the minimum pitch is made less, say 5 1/8 in., a 7/8-in. stay-bolt (Table 6) will answer the purpose, being good for just 100 lb. Or, with both pitches equal and below 5 3/8 in., a 7/8-in. stay-bolt pitched 5 1/4 x 5 1/4 in. will be good for 100 lb. pressure.

Consider another problem in which the pressure is 180 lb. gage and the plate 1/2 in.; find the pitch and diameter of the stay-bolt.

The greatest pitch permissible for this thickness of plate and pressure (see Table 1) will be 5 7/8 in. If the other pitch were made the same, a 1 1/4-in. stay-bolt would be good for 187 lb. This is more than necessary; hence using a pitch of 5 3/8 x 5 7/8 in., a 1 1/16-in. stay-bolt will answer, being good for 181 lb. If it is desired to use equal pitches, yet keep below 5 7/8 in., a 1 1/16-in. stay-bolt pitched 5 5/8 x 5 5/8 in. will be good for 180 lb. Of course, this increases the value of the plate as the maximum pitch is 5 5/8 in. and is good for 208 lb.

The tables are much simpler from the inspector's point of view. For example, let the pitches be 6x5 5/8 in., the stay-bolt 1 1/8 in. and the plate 3/8 in. thick. The maximum pitch is 6 in., and this is good for 107 lb. for a 3/8-in. plate (Table 1). A 1 1/8-in. stay-bolt pitched 6x5 5/8 in. at 6500 lb. stress will be found good for 148 lb. pressure. Therefore, the plate is the weaker and only 107 lb. should be allowed.

Again, let the plate be 7/16 in. thick, the stay-bolt 1 in. and the pitch be 5x4 3/4 in., the boiler being under United States Government jurisdiction. The maximum pitch is 5 in. and is good for 215 lb. for a 7/16-in. plate (Table 2). A 1-in. stay-bolt pitched 5x4 3/4 in. is good for 161 lb. at 6500 lb. (Table 7), but as the boiler is under government jurisdiction the bolts must be figured at 6000 lb. stress. The constant for this stress is 0.922. Therefore, the pressure would be

$$0.922 \times 161 = 148 \text{ lb.}$$

Hence, the stay-bolt is weaker than the plate and only 148 lb. should be allowed.

CURVES

Plotting the values given in the tables a set of curves was obtained as in Fig. 6. The curves for allowable pressure on plates according to the Massachusetts and United States Government formulas show at a glance the comparative values for a given pitch. Those based on Clark's formula do not show this as clearly as it must be remembered that what is termed pitch in the other formulas is "distance

Pitch	1" Stay-bolt Allowable Pressures at 6500 lb. per Sq. In. Stress																																	Area of Bolt at Root of Threads = 0.575 Sq. In. 12 Threads per Inch
	8	7 7/8	7 3/4	7 5/8	7 1/2	7 3/8	7 1/4	7 1/8	7	6 7/8	6 3/4	6 5/8	6 1/2	6 3/8	6 1/4	6 1/8	6	5 7/8	5 3/4	5 5/8	5 1/2	5 3/8	5 1/4	5 1/8	5	4 7/8	4 3/4	4 5/8	4 1/2	4 3/8	4 1/4	4 1/8	4	
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Pitch	1 1/16" Stay-bolt Allowable Pressures at 6500 lb. per Sq. In. Stress																																	Area of Bolt at Root of Threads = 0.662 Sq. In. 12 Threads per Inch
	8	7 7/8	7 3/4	7 5/8	7 1/2	7 3/8	7 1/4	7 1/8	7	6 7/8	6 3/4	6 5/8	6 1/2	6 3/8	6 1/4	6 1/8	6	5 7/8	5 3/4	5 5/8	5 1/2	5 3/8	5 1/4	5 1/8	5	4 7/8	4 3/4	4 5/8	4 1/2	4 3/8	4 1/4	4 1/8	4	
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TABLE 7. ALLOWABLE PRESSURES ON 1- AND 1 1/16-IN. STAY-BOLTS

in the clear" in Clark's formula. In other words, to get the pressure from the curves first subtract from the pitch the diameter of the stay-bolt, at the root of the thread.

It will be noted that the pressures allowed for ⅝-in. plate and 4-in. pitch are very great, in fact, 798 lb. by the United States Government and 750 lb. by Massachusetts. It can be shown mathematically that with a pitch of 2.449 in. the pressure allowed by the Massachusetts formula on ⅝-in. plates is *infinite.*

It should be stated, however, that the Massachusetts rules do not allow using ¼-in. plates for stayed flat surfaces. This plate, however, has been used in the calculations for comparison.

Curves are also plotted in Fig. 7, showing the allowable pressure on different size stay-bolts, for any area to be supported, at a stress allowance of 6500 lb.

There is another way in which a stay-bolt can fail, namely, by bending. In fact, this is very common, being caused by the unequal expansion of the outer sheets and furnace sheets. However, this is another problem and one about which very little can be predetermined.

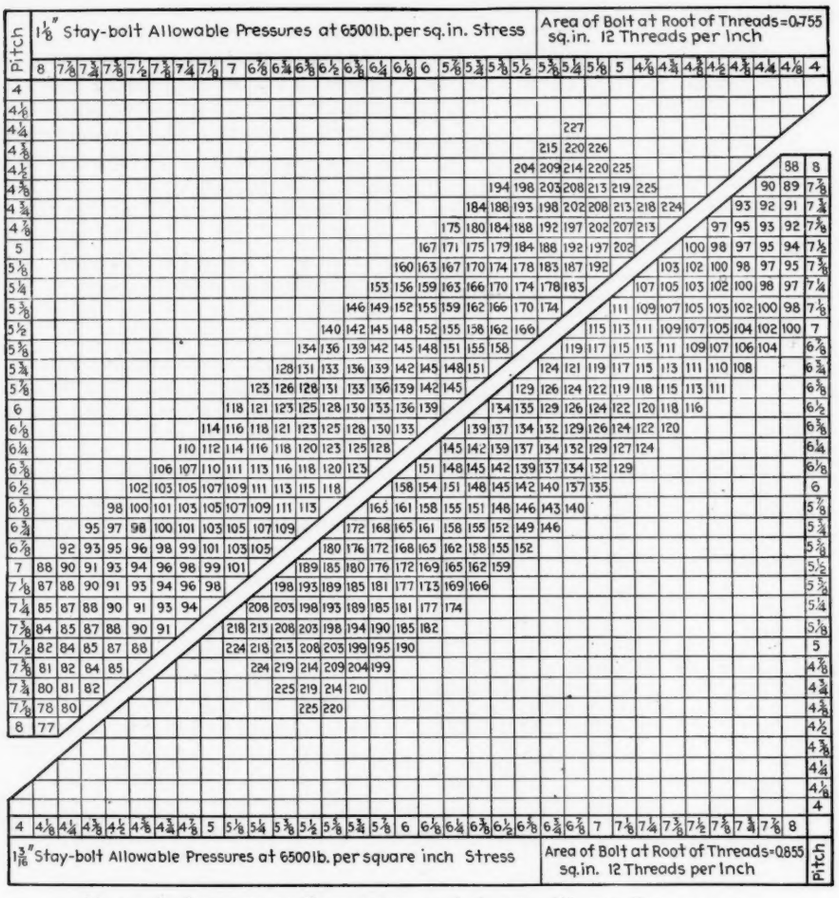

1⅛" Stay-bolt Allowable Pressures at 6500 lb. per sq. in. Stress
Area of Bolt at Root of Threads=0.755 sq. in. 12 Threads per Inch
Pitch
1 3/16" Stay-bolt Allowable Pressures at 6500 lb. per square inch Stress
Area of Bolt at Root of Threads=0.855 sq. in. 12 Threads per Inch

TABLE 8. ALLOWABLE PRESSURES ON 1⅛- AND 1 3/16-IN. STAY-BOLTS

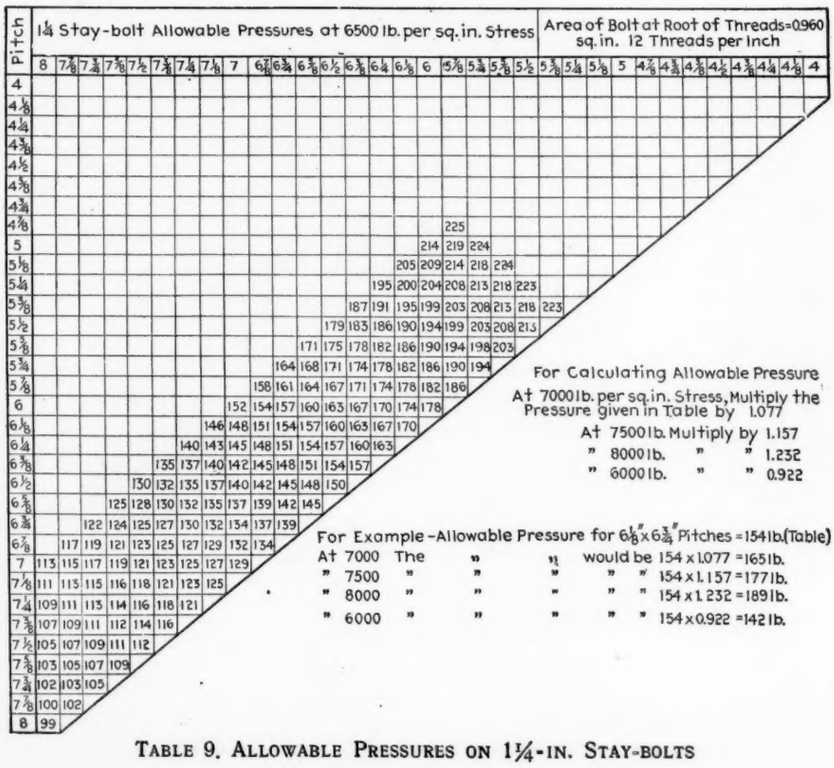

1¼ Stay-bolt Allowable Pressures at 6500 lb. per sq. in. Stress
Area of Bolt at Root of Threads=0.960 sq. in. 12 Threads per Inch
Pitch
For Calculating Allowable Pressure
At 7000 lb. per sq. in. Stress, Multiply the Pressure given in Table by 1.077
At 7500 lb. Multiply by 1.157
" 8000 lb. " " 1.232
" 6000 lb. " " 0.922
For Example - Allowable Pressure for 6⅛" x 6⅜" Pitches = 154 lb. (Table)
At 7000 The " " would be 154 x 1.077 = 165 lb.
" 7500 " " " " " 154 x 1.157 = 177 lb.
" 8000 " " " " " 154 x 1.232 = 189 lb.
" 6000 " " " " " 154 x 0.922 = 142 lb.

TABLE 9. ALLOWABLE PRESSURES ON 1¼-IN. STAY-BOLTS

Motor Driven Pipe-Threading Machines

A compact motive-driven pipe-cutting and threading machine has been designed by the Curtis & Curtis Co., 85 Garden St., Bridgeport, Conn., which cuts off and threads 2½- to 12-in. pipe.

The die-cutting head is of the Forbes pattern, and is mounted on an inclosed cabinet. The machine is self-contained and can be moved from place to place; after connecting the lead wires it is ready to operate. The various speeds obtainable and the starting and stopping of the machine are controlled by a lever, but the motor runs continuously except when the machine is shut down for long periods.

Thumbscrews for adjusting the dies are done away with as the dies are now clamped in position by one movement of a lever.

The motor is inside the base, where it is protected from the drippings of oil used in the screw-cutting operations. The machine can be taken from its base and used as a hand machine.

Huge Hydroelectric Project

It is reported that financiers have asked permission of the Norwegian government to harness the Aura and Lilledal Rivers, east of Molde, in the Romsdalen district, Norway. The scheme involves the erection of a dam 140 ft. high and the transmission of water through a tunnel to Sundalen, where 200,000 hp. could be developed.

The estimated cost of the project is $10,000,000 and the scheme if carried out will result in one of the largest hydro-electric power plants in the world. —*Electrical World.*

Water-power reserves were created by the Secretary of the Interior during October in California, Idaho, Montana, Oregon and Washington, covering an area of 36,902 acres.

Some Notes on the Slide Valve

By L. Holder

Why lap and lead are used on the valve and the effect they have on the events of the stroke. Diagrams to show the relative positions of eccentric and valve and piston and crank.

The simplest form of valve with which engineers have to do is the slide valve. It has four edges, two steam and two exhaust, and may be designed with or without lap. A lapless valve is so designed that it will just cover the port openings when the valve is at mid-travel, as shown in Fig. 1.

A valve may have both outside and inside lap. Outside lap means that the lips project beyond the port openings when the valve is at mid-position, as shown in Fig. 2. When a valve has inside lap its arch is shorter than the width of the exhaust port and the bridges,

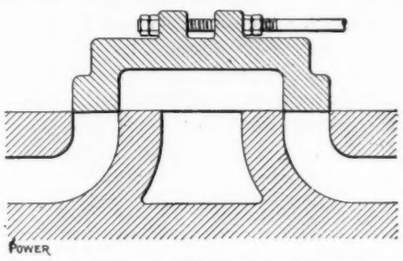


FIG. 1. LAPLESS VALVE IN CENTRAL POSITION

that is, the exhaust edges of the valve lap over the inside edges of the steam ports. This is shown in Fig. 3.

After examining these illustrations it will be evident that if there are two valve seats having equal steam ports and bridges (the thickness of metal between the steam and exhaust ports) and if one has a wider exhaust port than the other, the former will require a correspondingly wider arch, but there will be no difference in the travel, lap or point of cutoff. Supposing, however, that these same valve seats have equal steam and exhaust ports but a difference in the width of the bridge, the one having the wider bridge will require a proportionately longer arch. There will be no difference in the lap or point of cutoff, but to give free admission the wider bridge will require a longer travel. Again, if the two valve seats have the same size of exhaust ports and bridges but have different widths of steam port, the valve having the wider steam ports will require proportionately wider laps.

Most engineers know that a valve with lead opens a little before the engine is on dead center. With negative lead the valve does not uncover the port until the piston has started on its return stroke. In other words, the valve is set with late admission. Inside lap has nothing to do with steam admission nor does it affect the point of cutoff, but it does increase and hastens compression and retards the exhaust release, or, in other words, it tends to choke the escape of the exhaust steam. It allows expansion of the steam to take place during a longer period than if there were no inside lap. On the other hand, increasing the outside lap has no effect on the compression in the cylinder and it neither hastens nor retards the release of the exhaust steam, but the more outside lap a valve has the later the admission will take place.

Suppose the cutoff is retarded, due to increased travel of the valve, it may be neutralized by increasing the outside lap of the valve. Late compression can be corrected by adding more inside lap to the valve. If the exhaust takes place too late, due to excessive travel of the valve, the inside lap can be reduced or entirely removed.

When a valve has neither lap nor lead

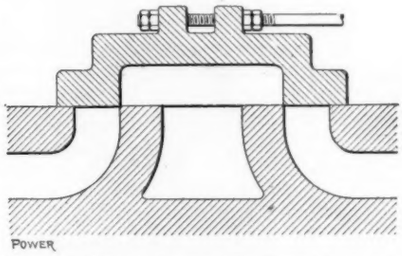


FIG. 2. VALVE WITH OUTSIDE LAP IN CENTRAL POSITION

the eccentric should be set at right angles to the crank and ahead of it. If there is lead and lap the eccentric should be set enough more than 90 deg. ahead of the crank to move the valve the length of the lap. Thus it will be seen that the position of the valve will depend upon the position of the eccentric. On some engines the eccentric is fixed, but on others its position on the shaft may be changed so as to vary the position of the valve during the stroke. This regulates the cutoff, so that just the proper amount of steam is admitted to carry the load. The governor changes the position of the eccentric and the eccentric governs the travel of the valve. Such an engine is of the variable-cutoff type. When equipped with a single valve the outside and inside lap is fixed, and the governor varies the cutoff by changing the valve travel or angular advance, and sometimes both.

To many engineers the effect of changing the movement of the valve is not understood. For instance, if the outside lap is increased the admission of steam begins later and ceases sooner, and the expansion of the steam occurs earlier and continues longer; the exhaust and compression remain the same. If the outside lap is decreased the admission of steam begins earlier and ceases later, while the expansion begins later and the period shortens, the exhaust and compression remaining the same.

If the inside lap is increased, the steam admission will not be changed, the expansion will begin as before and continue longer, the exhaust will begin later and cease earlier, while the compression will begin sooner and continue longer. If the inside lap is decreased the steam admission remains unchanged, the expansion will begin as before but the period of expansion will be shortened, the exhaust will begin earlier and cease later, and the compression will begin later so that the period will be shortened.

This is not all, for if the valve travel is increased the admission of steam will begin sooner and cease later, the expansion will begin later and cease sooner. The exhaust will begin earlier and cease later and the compression will begin later and end sooner. If the valve travel is decreased the admission of steam begins later and ceases earlier, expansion begins earlier and ceases later. The exhaust and compression is also changed, for the exhaust begins later and ceases earlier while the compression begins earlier and ceases later.

Moving the eccentric around on the engine shaft changes the angular advance of the valve but does not vary the travel of the valve. The events of the stroke all occur earlier but the periods remain unchanged. If the angular advance is decreased the events begin later but the periods are not changed.

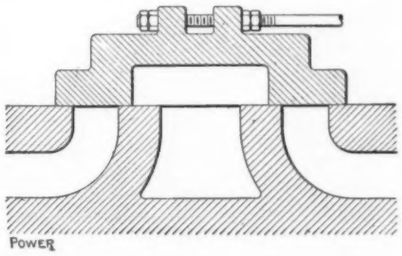


FIG. 3. VALVE WITH BOTH OUTSIDE AND INSIDE LAP

The lead varies with different engines and is given for the purpose of filling the clearance space of the cylinder with live steam before the piston begins the stroke. High-speed engines require more lead than slow-speed engines. The purpose of lap is to give the steam an oppor-

tunity to expand in the cylinder after cut-off occurs, and to keep the ports covered during the time expansion is taking place both inside and outside lap is given to the valve.

The position of the valve for corresponding positions of the eccentric can be determined by the use of a diagram such as Fig. 4. First draw a circle representing the path of the eccentric during one revolution of the engine, the

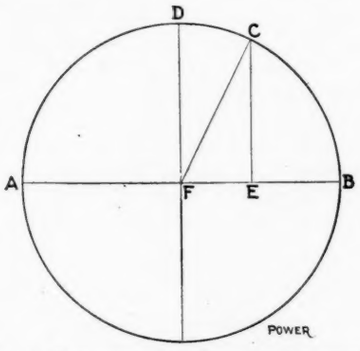


FIG. 4. RELATIVE POSITION OF ECCENTRIC AND VALVE

diameter *AB* indicating the valve travel. Next designate a point on the circumference of the circle which will represent the center of the eccentric, as at *C*. It is then an easy matter to find the distance the valve has moved from mid-position by drawing a perpendicular line from the point *C* to the line *AB*. The valve has moved the distance *FE* from mid-position.

The angularity of the eccentric is small and may be neglected, but is greater for the connecting-rod, so that the displacement of the piston relative to the crank must be found by the use of a diagram such as is shown in Fig. 5. In this instance the line *OH* represents the stroke, *J* the piston, *K* the crosshead, *KL* the connecting-rod and *LM* the crank. With the parts in the position indicated in the

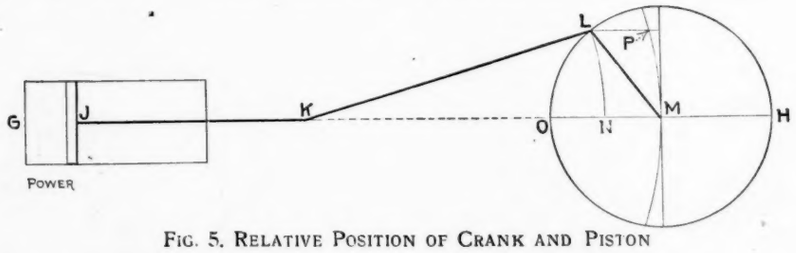


FIG. 5. RELATIVE POSITION OF CRANK AND PISTON

illustration, if the connecting-rod were of infinite length the displacement from the end of the cylinder would be equal to the distance from *O* to a line from *L* perpendicular to *OH*. To find the true displacement draw the arc of a circle *LN* through *L*, having its center at *K*. Then *NM* equals the displacement from the middle of the cylinder or *ON* equals the displacement from the end.

The same results could be obtained by drawing an arc having the same radius through *M* instead of through *L*, and by drawing the line *LP* parallel with *OH*, in which case *LP* being equal to *NM* indicates the displacement from the center of the cylinder.

Blonck Boiler Efficiency Meter

The Blonck boiler-efficiency meter illustrated herewith, consists essentially of a differential gage, showing the drop in draft through the boiler and a direct gage showing the draft in the furnace. The differential gage is a measure of the volume of gas passing through the boiler. The direct gage is a measure of the resistance of the fuel bed.

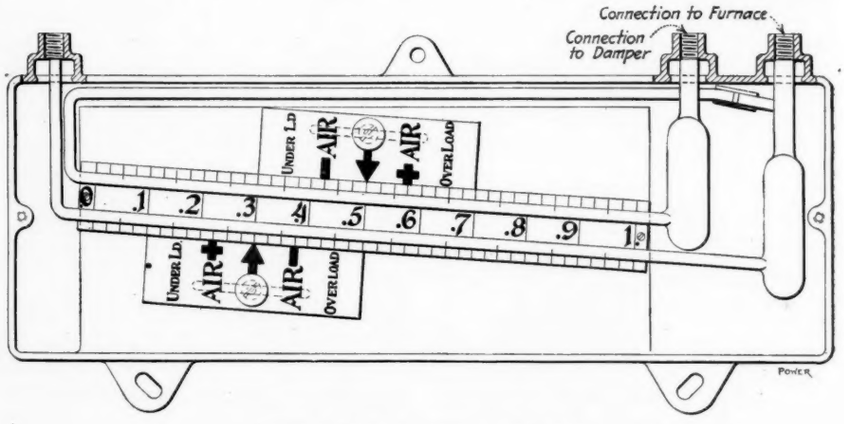
For a given boiler, a given grade of fuel and given rate of working there is a certain correct drop in draft between the furnace and the damper. Assuming a newly cleaned fire, under normal working conditions the resistance to the air passing through the fuel will be at the minimum and it will gradually increase as the thickness of the fuel bed increases, due to the accumulation of ashes on the grate, until just before the next cleaning, then it will be at the maximum. Each time fresh fuel is thrown on the fire this resistance to the passage of air increases slightly, but as this fuel ignites and burns the resistance gradually becomes less until the draft is once more the same or very nearly the same as it was before the firing.

The drop in draft through the boiler should remain constant as long as the rate of steaming remains the same and the damper should be adjusted to keep it so.

Thus, if the differential gage reads low, indicating an insufficient flow of air, the damper must be opened slightly to bring the amount up to normal. The furnace draft will also show an increase, indicating that the thickness of the fuel bed has increased.

If the differential draft is high, too much air is being admitted. The furnace draft may be low at this time, indicating that the fire is too thin. Or, the furnace draft may be normal, indicating that the damper in the stack should be throttled to reduce the effective draft in the boiler.

The important advantages of this instrument are its simplicity and its rapidity of action. The fireman can be quickly taught what the changes of draft mean and what should be done to bring conditions back to normal. Its rapidity of action makes it possible to instantly detect wasteful conditions and to correct them quickly.



BLONCK BOILER-EFFICIENCY METER

This meter is manufactured by W. A. Blonck & Co., Fisher Building, Chicago, Ill.

The resuscitation-from-shock chart recently circulated by the National Electric Light Association will be adopted by the Navy Department, and policemen and firemen in large cities are making good use of it. It has a wide use among lighting companies, one Middle Western company alone having ordered 1000 copies for distribution among its power-plant and other employees.

The effect of vanadium in plain carbon-steel castings is to increase the elastic limit about 30 per cent., giving a much higher proportion of available strength for the same ultimate strength. Moreover, it increases the power of castings of this character to withstand repeated and alternating stresses fully 50 per cent.

Rotary Jet Condenser

A new type of rotary jet condenser and high-vacuum, rotary air pump, has recently been placed on the market. It is a development of the Rees "Roturbo" type of centrifugal pump, described in POWER, Sept. 10, 1912, the feature of which was the employment of a revolving accumulator or pressure chamber. In the condenser, this pressure chamber is used to lift the condensing water to a limit of 20 or 25 ft., and give it an initial pressure after raising it.

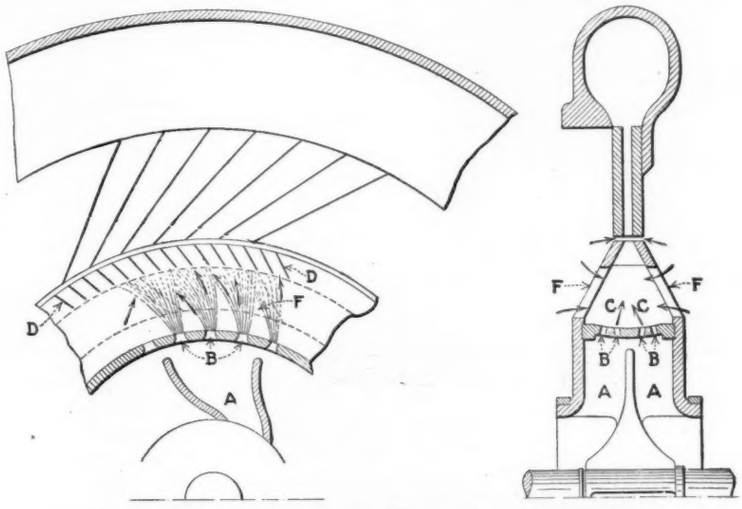


FIG. 1. PARTIAL SECTIONS OF ROTARY JET CONDENSER

Water flows up the suction pipe into this revolving pressure chamber *A*, Fig. 1, on the periphery of which nozzles *B* are arranged, not less than ⅜ or ½ in. in diameter, on the smaller, and up to ¾ or 1 in., on the larger sizes. These nozzles allow the water to escape from the pressure drum in radiating jets which are arranged to impinge in pairs, making the water jets fan-shaped and subdividing the water into a fine spray, that is projected in lines radiating from the shaft, but still rotating as a whole with the impeller, across a space *C*, into which steam exhausts, through the openings *F*. This, it is said, ensures a perfect mixture of steam and water which is ideal for the most rapid condensation.

Air entrained with the steam and the condensate are picked up by the blades of the exhausting fan *D*. Fundamentally these rim blades are the equivalent of the mixing cone, and the neck or "shock zone" of the old ejector, adapted to be rotated to give a positive exhausting effect through centrifugal action.

The neck of the stationary ejector is the seat of most of the losses due to shock, or the sudden change of velocity of the water from the high speed at which it is projected across the vacuum chamber to the much lower velocity of the mixture of air and water in the neck, where the initial compression of the air takes place, and where the volume of air is suddenly reduced. The mechanical ejection of the condensate in this condenser and the initial compression of the air by the fan blades removes the old difficulties of choking and gives high efficiency with certainty of ejection.

These exhausting fan blades are said to also form an automatic regulator, which deals with the fluctuating conditions of steam or vacuum without any speed regulation. For instance, if the vacuum is reduced below the normal, the rim blades are practically empty, and the condensate is swept out through the exhausting fan with but little assistance from the blades. If the vacuum rises beyond normal, thus exerting a greater backward pull, tending to cause choking, the rim blades fill up with water to a greater depth. Because of the extra centrifugal force created with a greater depth of the water carried around by the fan blades, a greater exhausting effect is secured when running at the same speed.

This regulating effect is absent on other devices of this kind, it is claimed, because the water has not heretofore been controlled by the rotary member at both the point of injection to, and ejection from the vacuum chamber, and at the same time spraying freely across a large intervening condensing space.

This feature, allowing for long, impinging jets, which form the finest spray and a great area for condensation, enables the apparatus to be used for the largest size condensers, without any auxiliary apparatus for extracting the air or water.

In Fig. 2 is shown an outside view of a motor-driven unit designed for condensing 30,000 lb. of steam per hour at 28 in. vacuum. The revolving impeller is fitted to the shaft as shown in Fig. 3. Fig. 4 shows the impeller with the side plate removed. A sectional view of the condenser is given in Figs. 5 and 6. When run as an air pump the same design of impeller is used and practically the only difference is that the external casing is of a different shape and has a smaller opening for the entrance of

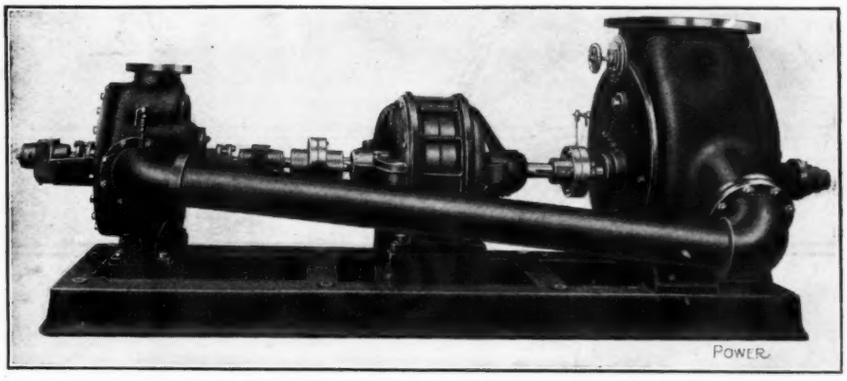


FIG. 2. MOTOR-DRIVEN ROTARY JET CONDENSER

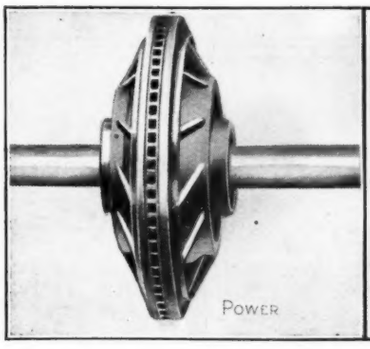


FIG. 3. IMPELLER OF ROTARY CONDENSER

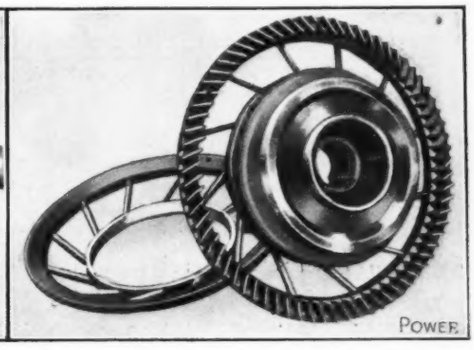


FIG. 4. IMPELLER WITH SIDE PLATES REMOVED

the air than would be required for steam.

Low first cost and maintenance, small floor space required and simplicity of construction are other advantages claimed.

Fig. 7 shows performance curves from manufacturer's tests of this machine, with various volumes of steam, etc. An allowance was made for 3 cu.ft. of free air per minute per 10,000 lb. of steam condensed per hour, for reciprocating engines and half of this for turbines.

Suppose it is desired to find the horsepower required by the condenser and the amount of cooling water at 60 deg. F. for a steam turbine using 30,000 lb. of steam per hour, and exhausting into a vacuum of 28 in.

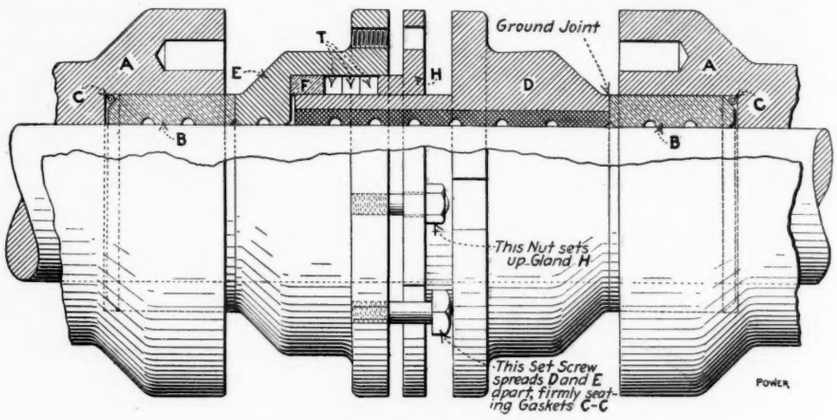
On the left side of the diagram run up the vertical line above 28 in. until the curve marked 60 deg. is reached; then follow this line horizontally to the right until it intersects the vertical line from 30,000 lb. of steam per hour. This point falls on or very near the curve for 40-hp. and 120,000 gal. of water per hour. The curves in Fig. 8 are for the air pump only, and are used somewhat similarly.

The Manistee Iron Works, Manistee, Mich., has the United States rights in this British invention.

Metal Pump Plunger Packing

This packing is a continuous connecting sleeve between two cylinders, and closely fits the pump plunger. It has water grooves to cause a reversal of flow of any fluid that may tend to pass the sleeve. The gland shown herewith is for a slip joint, and no wearing from the plungers occurs to the square hydraulic packing *T*, which is a permanent fixture, to be renewed only in case of accident. The packing is held in place by the gland *H*. *F* is a bottom ring. This slip joint fits between the parts *D* and *E*. The two parts may be forced apart after being placed between the cylinders to seat the gaskets *C* at the bottom of the stuffing-boxes *A*, which receive the bushings. On most outside-packed pumps, the sleeve may be made from 12 to 18 in. long. When wear occurs *D* is relined.

This packing is made by the Metallic Packing & Mfg. Co., Elyria, Ohio.



METAL PUMP-PLUNGER PACKING

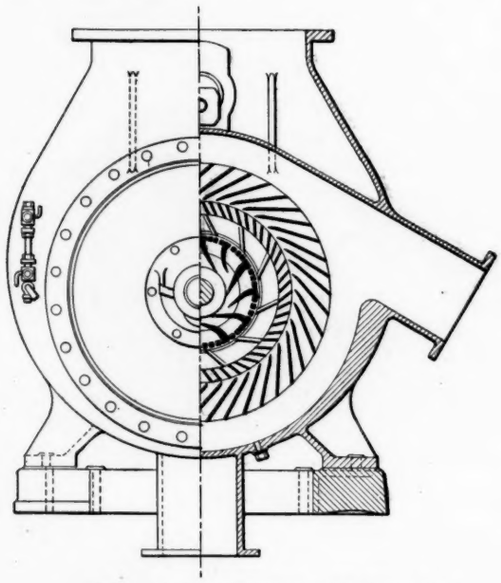


FIG. 5. SEMI-SECTIONAL END VIEW OF ROTARY JET CONDENSER

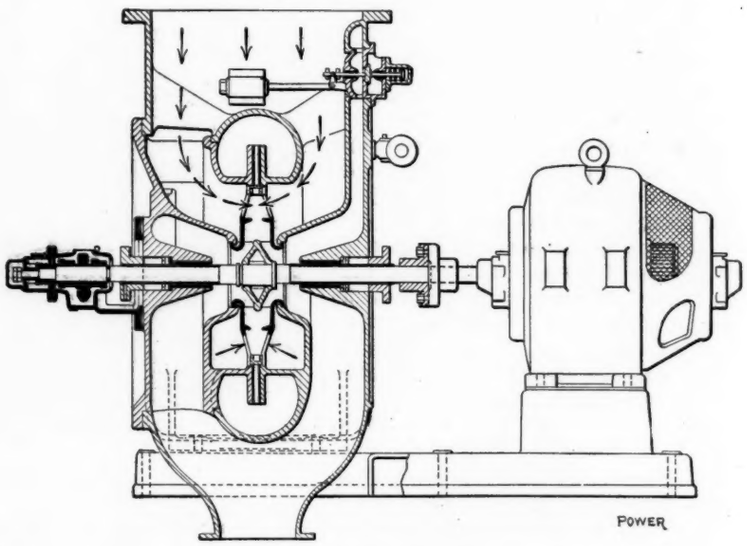


FIG. 6. SECTION THROUGH ROTARY JET CONDENSER

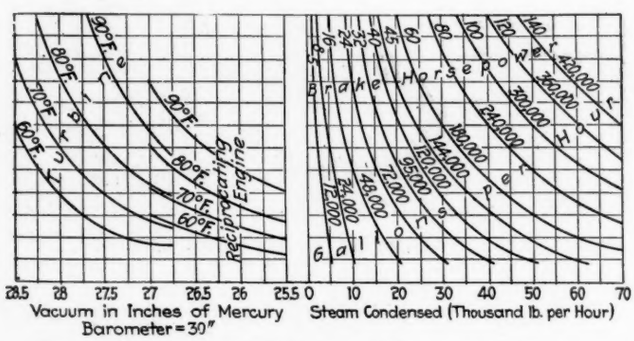


FIG. 7. CURVES SHOWING VARYING VOLUME OF STEAM AND TEMPERATURE OF COOLING WATER

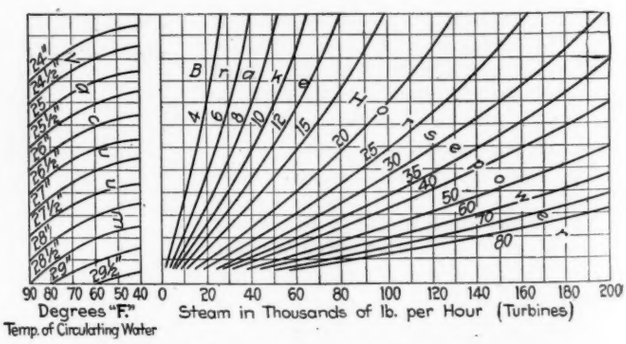

Brake Horsepower
Vacuum
90 80 70 60 50 40
Degrees "F."
Temp. of Circulating Water
0 20 40 60 80 100 120 140 160 180 200
Steam in Thousands of lb. per Hour (Turbines)

FIG. 8. PERFORMANCE CURVES OF ROTARY AIR PUMP

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

Primer of Electricity

By Cecil P. Poole

The Series-wound Motor

This type of machine is used only for work requiring a heavy torque at starting and where large speed changes are not objectionable, as in the cases of railways, cranes, hoists and similar classes of service. Series-wound motors are supplied from constant-potential circuits almost exclusively and, as a rule, the entire armature current passes through the field windings. Any change in load, therefore, causes a change in speed much greater than that of a shunt-wound motor. When the current increases, it not only produces an increase in the voltage drop through the resistance of the windings but strengthens the field magnet and thereby increases the counter electromotive force of the armature for a given speed.

Because of the changes of field strength with changes in load, the speed of a series-wound motor is very unstable. Such a machine cannot be left in circuit without a load because the armature would race to such an excessive speed as to destroy itself by centrifugal force.

The series-wound motor possesses the great advantage, however, of enormous starting torque—hence its use for railway cars and hoists. The reason for this is that torque is directly proportional to both the field-magnet flux and the armature current, and, as already explained, the armature current is also the field current; therefore, a heavy armature current produces a heavy field-magnet flux. The field-magnet flux does not vary in direct proportion to variations in the motor current, because the field-magnet cores and armature teeth are magnetized considerably above the bend of the "saturation" curve.* Doubling the current, therefore, does not double the magnetic flux.

Fig. 116, which is the excitation curve of a series-wound motor, illustrates the effect of current changes upon the field strength of the machine. With 10 amp. in the field winding the total flux in the air gaps between the pole faces and the armature core was 2,000,000 maxwells; with an increase to 20 amp. the flux increased to only 3,800,000 maxwells, and four times this exciting current did not quite double the flux.

The test data from this machine are presented in Table 7, and a careful study of these figures should be made in order to get a grasp of the characteristic behavior of series-wound motors. The numbers in the extreme right-hand column

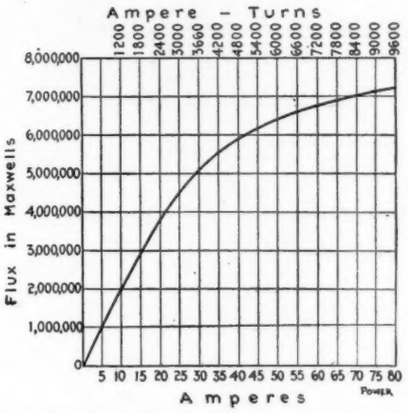


Fig. 116. Excitation Curve of a Series-wound Motor

are total horsepower developed by the armature; that is, the external power plus the power absorbed in revolving the armature and supplying core losses. This corresponds precisely with the indicated horsepower of a steam engine, and the power lost in the windings and connections corresponds very closely with the loss due to condensation in an engine.

TABLE 7. TEST DATA FROM A 20-HP. SERIES-WOUND MOTOR

250 volts. 800 armature conductors. 2 armature paths. 120 turns in each field-magnet coil. Total resistance of armature and field-magnet windings, 0.4 ohm.

Amperes	Ampere Turns	Total Flux	Total Drop	Counter E.m.f.	R.p.m.	Torque	Total Hp.
10	1200	2,000,000	6	244	1830	9.38	3.27
15	1800	2,950,000	8	242	1260	20.8	4.87
20	2400	3,800,000	10	240	947	35.7	6.43
25	3000	4,540,000	12	238	786	53.3	8.0
30	3600	5,100,000	14	236	694	71.9	9.5
35	4200	5,550,000	16	234	632	91.2	11.0
40	4800	5,900,000	18	232	590	110.8	12.4
45	5400	6,170,000	20	230	559	130.3	13.9
50	6000	6,400,000	22	228	535	150.2	15.3
55	6600	6,575,000	24	226	516	169.7	16.7
60	7200	6,740,000	26	224	498	189.8	18.0
65	7800	6,880,000	28	222	484	209.9	19.3
70	8400	7,000,000	30	220	471	230.0	20.6
75	9000	7,100,000	32	218	460	250.3	21.9
80	9600	7,200,000	34	216	450	270.3	23.2

In fact, a series-wound motor operating on a constant-potential circuit has almost all of the general characteristics of a steam engine supplied at constant pressure and running without a governor.

Referring again to Table 7, it will be found that the drop due to the electrical resistance of the machine is given as being 2 volts greater than the product of *Resistance* × *Current*. The explanation of this is that the drop at the brush faces was almost constant at 2 volts for all loads, the lowest drop being 1.9 volts and the highest 2.15 volts. (See lesson of July 26, 1910, on the resistance of brush contacts.) The table shows very clearly how rapidly the torque increases with the current and how the horsepower increases almost in proportion to the current, in spite of the great reduction in speed.

A chart made by plotting these figures in the form of curves, however, gives a clearer grasp of their significance. Fig. 117 is such a chart. The "R.p.m." curve, for instance, is much more effective than the tabulated figures in conveying to the mind the rapid decrease of speed during the first part of the range. It also illustrates the statement regarding the tendency of a series-wound motor to race at light loads; if the left-hand end of the curve were prolonged it would show a speed of about 3700 r.p.m. at the 5-amp. line of the current scale.

A series-wound motor is usually provided with a starting rheostat constructed on the same general principle as that of the shunt-wound machine. As the field winding is in series with the armature, the rheostat is connected in series with the whole machine, as indicated by Fig. 118. In some cases, the connections are arranged so that when the contact arm is at the end of its travel, giving maximum speed, the resistor that has been cut out of the main circuit is connected in parallel with the field winding. This "shunts" some of the current out of the winding and weakens the field, thereby compelling the armature to run at a higher speed in order to generate the counter electromotive force required to balance the impressed electromotive

*See lesson on Dynamo Field Excitation, Nov. 8, 1910.

force. Fig. 118 illustrates the electrical principle of such a rheostat but not the usual mechanical construction. When the arm first reaches the last contact, the resistor *Rh* is cut out of the circuit; further movement of the arm to the right will bring the auxiliary finger *F* into contact with the first button, thereby connecting the otherwise free end of the resistor to the far end of the field winding and putting the two in parallel. Besides taking some of the current out of the field winding, this reduces the electrical resistance of the circuit and therefore the drop due to resistance. The required counter electromotive force is thereby increased, entailing still more increase in speed.

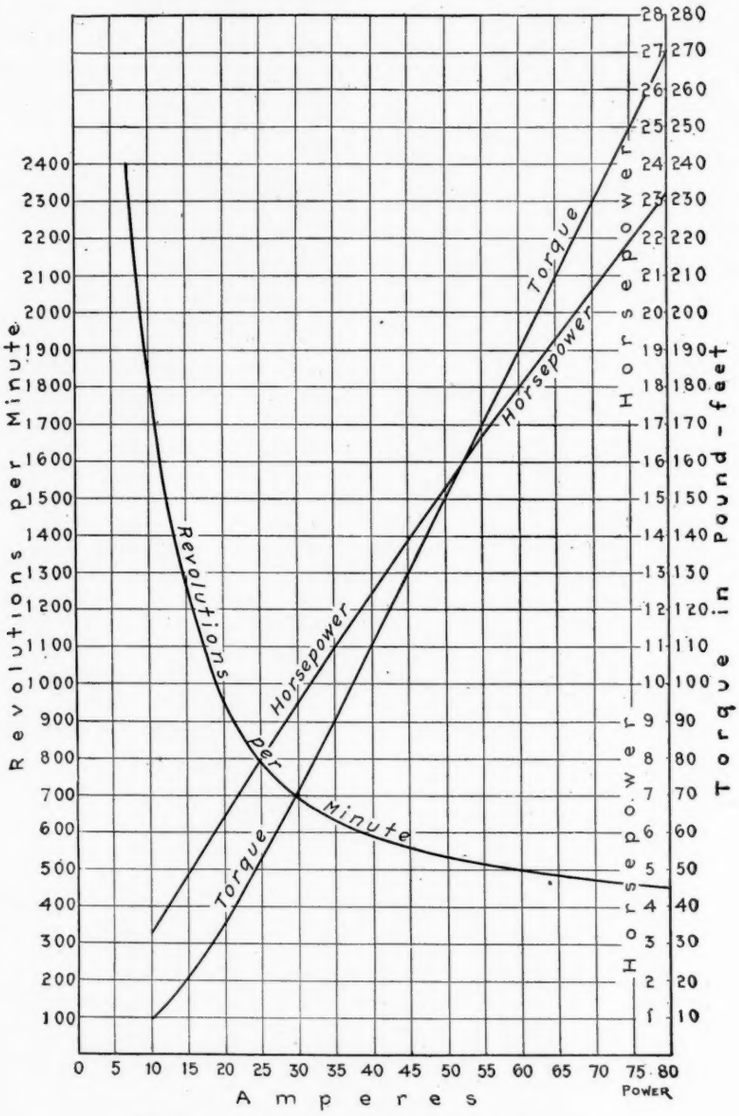


FIG. 117. CHART OF SERIES MOTOR PERFORMANCE

THE COMPOUND-WOUND MOTOR

To avoid the excessive speed changes of the series-wound motor and at the same time obtain the advantage of a powerful starting torque, motors are built with "cumulative" compound windings. (See Fig. 111.) The series winding gives the heavy torque at starting and the shunt winding provides a steady field flux which keeps the armature from running away when the load is not heavy enough to hold the speed down with a simple series winding.

For example, suppose the motor last discussed had been equipped with a shunt winding of 7000 ampere-turns per pole, which produced a total flux of 6,690,000 maxwells, and a series winding of 20 turns per pole. For convenient comparison also assume the total resistance of the armature and series field windings to be 0.4 ohm, as before. With the full load of 20 hp. the armature current of 70 amp. would produce

$$70 \times 20 = 1400 \textit{ ampere-turns}$$

in the series winding and these added to the 7000 shunt ampere-turns would make the total excitation 8400 ampere-turns. This would produce the normal total flux of 7,000,000 maxwells (see Table 7) and the speed would be 471 r.p.m., as in the case of the series-wound machine. But if the load were removed entirely, leaving the series winding practically dead, the shunt winding would maintain 6,690,000 maxwells. The drop in the armature circuit would be at least 3 volts, leaving 247 volts to be opposed by counter electromotive force; the speed, therefore, would not rise above

$$\frac{6{,}000{,}000{,}000 \times 2 \times 247}{6{,}690{,}000 \times 800} = 554 \textit{ r.p.m.}$$

With the series-wound machine previously described, a decrease of only 6 hp. in the load would produce the same increase in speed.

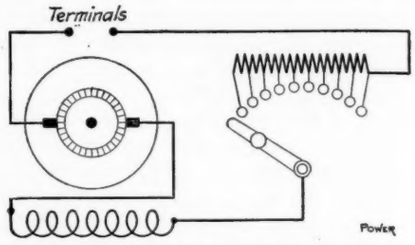


FIG. 118. SERIES-WOUND MOTOR AND STARTING RHEOSTAT

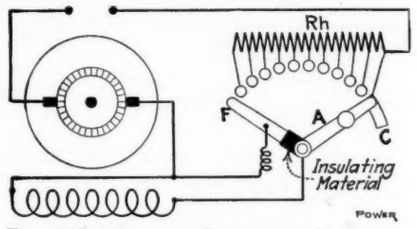


FIG. 119. SPECIAL RHEOSTAT SHUNTING FIELD WINDING FOR MAXIMUM SPEED

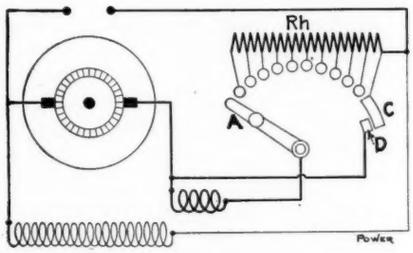


FIG. 120. COMPOUND-WOUND MOTOR AND STARTING RHEOSTAT

As the preceding example shows, the speed regulation of a compound-wound motor is poor. When a fairly close approach to steady speed is desired, therefore, together with powerful starting torque, a compound-wound motor is used and with it a starting rheostat arranged to cut out the series winding after the resistor has been cut out. A diagram of such an arrangement is shown in Fig. 120. The final contact *C* of the rheostat is a curved plate instead of a button, and an auxiliary contact *D* is located near the end of it so that the arm *A* will touch both plates at the end of its travel. As the arm is connected to one terminal of the series field winding and the contact *D*

to the other, the winding is short-circuited when the arm rests on the contact.

Compound-wound motors are not commonly built with differential field windings now. This type was used to some extent years ago when the speed regulation of the shunt-wound motor was not good. The effect of the differential winding is to weaken the field strength as the load increases, thereby causing the speed to be higher than if the series winding were not there.

For example, take the machine just considered, but assume that it is equipped with a shunt field winding containing 8400 ampere-turns per pole, producing a flux of 7,000,000 maxwells, and suppose the series winding to contain 40 turns per pole and to be connected to oppose the shunt winding. Assume the resistance of the armature circuit to be 0.4 ohm, as before. At no load the effect of the series winding would be too small to consider, and (with an armature-circuit drop of 3 volts) the speed would be

$$\frac{6{,}000{,}000{,}000 \times 2 \times 247}{7{,}000{,}000 \times 800} = 529\ r.p.m.$$

With a load sufficient to cause 70 amp. to pass through the armature circuit, the series winding would have an exciting force of 2800 ampere-turns, and, this being opposed to the shunt winding of 8400 ampere-turns, there would be 5600 ampere-turns per pole left active. According to the excitation curve, Fig. 116, 5600 ampere-turns will produce a total flux of 6,240,000 maxwells. The speed, therefore, with this load would be

$$\frac{6{,}000{,}000{,}000 \times 2 \times 220}{6{,}240{,}000 \times 800} = 529\ r.p.m.$$

the same as at no load.

The speed of a differentially-wound motor, however, is not the same at all loads, because the excitation curve is not a straight line; that is, the change in flux is not proportional to the change in ampere-turns. If the winding be designed to give the same speed at no load and at full load, the speed at all loads between will be lower. If the winding gives the same speed at half load as at no load, the speed below half load will be lower and that at all loads above one-half will be higher than the no-load speed. A complete discussion of the differentially-wound motor is not worth while, because this type has gone almost out of use.

Conduit Sizes

Recognizing the tendency in the past to use conduits of too small diameter, the National Electrical Contractors' Association has prepared data as to the proper sizes of conduit to be used in installing wires and cables. As relates to power and lighting systems this is as given in the accompanying tables.

The conduit sizes are based on the use of not more than three 90-deg. elbows in runs taking up to and including No. 10 wires; and two elbows for wires larger than No. 10; wires No. 8 and larger are stranded.

THREE-WIRE CONVERTIBLE SYSTEM

Size of wires, B. & S. Gage		Size conduit, in.	
2-wire	1-wire	Internal	External
14	10	¾	1.05
12	8	¾	1.05
10	6	1	1.31
8	4	1	1.31
6	2	1¼	1.66
5	1	1¼	1.66
4	0	1½	1.9
3	00	1½	1.9
2	000	1½	1.9
1	0,000	2	2.37
0	250,000	2	2.37
00	350,000	2½	2.87
000	400,000	2½	2.87
0,000	550,000	3	3.5
250,000	600,000	3	3.5
300,000	800,000	3	3.5
400,000	1,000,000	3½	4
500,000	1,250,000	4	4
600,000	1,500,000	4	4.5
700,000	1,750,000	4½	5
800,000	2,000,000	4½	5

NUMBER OF WIRES IN SYSTEM

Size, B. & S. Gage	Capacity Amp.	One wire in a conduit Size of conduit, in.		Two wires in a conduit Size of conduit, in.		Three wires in a conduit Size of conduit, in.		Four wires in a conduit Size of conduit, in.	
		Internal	External	Internal	External	Internal	External	Internal	External
14	12	½	0.84	½	0.84	½	0.84	¾	1.05
12	17	½	0.84	¾	1.05	¾	1.05	¾	1.05
10	24	½	0.84	¾	1.05	¾	1.05	1	1.31
8	33	½	0.84	1	1.31	1	1.31	1	1.31
6	46	½	0.84	1	1.31	1¼	1.66	1¼	1.66
5	54	¾	1.05	1¼	1.66	1¼	1.66	1¼	1.66
4	65	¾	1.05	1¼	1.66	1¼	1.66	1½	1.9
3	76	¾	1.05	1¼	1.66	1¼	1.66	1½	1.9
2	90	¾	1.05	1¼	1.66	1½	1.9	1½	1.9
1	107	¾	1.05	1½	1.9	1½	1.9	2	2.37
0	127	1	1.31	1½	1.9	2	2.37	2	2.37
00	150	1	1.31	2	2.37	2	2.37	2½	2.87
000	177	1	1.31	2	2.37	2	2.37	2½	2.87
0,000	210	1¼	1.66	2	2.37	2½	2.87	2½	2.87
CM.									
250,000	235	1¼	1.66	2½	2.87	2½	2.87	3	3.5
300,000	270	1¼	1.66	2½	2.87	2½	2.87	3	3.5
400,000	330	1½	1.66	3	3.5	3	3.5	3½	4
500,000	390	1½	1.9	3	3.5	3	3.5	3½	4
600,000	450	1½	1.9	3	3.5	3½	4		
700,000	500	2	2.37	3½	4	3½	4		
800,000	550	2	2.37	3½	4	4	4.5		
900,000	600	2	2.37	3½	4	4	4.5		
1,000,000	650	2	2.37	4	4.5	4	4.5		
1,250,000	750	2½	2.87	4½	4.5	4½	5		
1,500,000	850	2½	2.87	4½	5	5	5.56		
1,750,000	950	3	3.5	5	5.56	5	5.56		
2,000,000	1050	3	3.5	5	5.56	6	6.62		
Duplex Wire.									
14	12	½	0.84	¾	1.05	1	1.31	1	1.31
12	17	½	0.84	¾	1.05	1	1.31	1¼	1.66
10	24	¾	1.05	1	1.31	1¼	1.66	1¼	1.66

This information has also been prepared in the form of a set of charts showing the conduit and conductors in full size. These have been copyrighted by the National Electrical Contractors' Association, Utica, N. Y., and are furnished by them at cost.

CORRESPONDENCE

Truing Commutators

If a commutator is kept properly trued by the use of sandpaper on a curved block or any other good smoothing device, it is hard to see why it should ever be necessary to take it to the turning lathe. However, if the grinding apparatus is not properly made and used it will grind more on one side than the other or grind in spots. In either case the trouble seems always to become worse with each grinding rather than better.

I always use two pieces of sand stone or grind rocks if I can get them. They should be chiseled out to fit, and each should reach nearly half-way around the commutator if possible. I use two pieces in order to be certain that the pressure is exactly the same on both sides and for the same reason always grind while the armature is revolving at a fair speed.

An attempt to smooth a commutator by drawing the sandpaper back and forth on it while it is not turning is quite certain to throw it out of round and the use of a short block or stone will usually grind low spots; but a stone reaching one-third or more around the commutator will touch only the high places and automatically keep it ground to a true circle.

Where the commutator is quite large I make the half circle with several stones clamped between two boards. The boards should be heavy and the bolts drawn up very tightly to prevent accident. The same stones may be used on two or more sides to fit the different circles and they may be rearranged in the clamp.

FRED BOONE.

Davidson, Okla.

A company has been organized in Rio de Janeiro, Brazil, for the purpose of utilizing the Paulo Alfonso Falls of the San Francisco River to generate electric power. At first 200,000 hp. will be generated. which quantity is to be increased at a later date to 300,000 hp.

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

Gas and Oil Power Development in Europe

One of the most instructive and comprehensive treatises on current European practice in gas and oil power development is contained in the report of H. I. K. Freyn, chairman of the Gas Power Section of the American Society of Mechanical Engineers. Mr. Freyn recently returned from a trip of five or six months abroad devoted almost exclusively to a study of gas and oil power conditions, and the results of his findings were given at the recent annual meeting of the society. His report is, in part as follows:

The Diesel Engine

Without desiring to detract from the importance of the Diesel engine my careful investigation has shown that the warning against over-enthusiasm which I sounded about a year ago on a similar occasion was fully warranted. In this connection I refer more particularly to the large Diesel engine, above 1000 hp. per unit, about which very optimistic reports and accounts appeared from time to time. The actual achievements do not altogether coincide with these glowing accounts nor with the claims of some manufacturers.

However, it is unnecessary to dwell on the importance of the Diesel engine for ship propulsion. Efforts are being made by many manufacturers of marine Diesel engines in Europe, to design an engine which, while giving nearly the same excellent fuel economy as the stationary engine, will also meet the severe requirements of simplicity, reliability and reduced first cost. That the last condition is not yet fulfilled is shown by the fact that according to authentic information the cost of an 800-hp., four-cylinder, two-stroke-cycle marine engine recently built was found to be 50 per cent. higher than anticipated and 150 per cent. higher than that of corresponding steam-engine equipment.

I was unable to find anywhere the 10,000 and 15,000-hp. Diesel-engine ships mentioned in current literature and even Diesel-engine enthusiasts do not believe that our modern leviathans will ever be propelled by Diesel engines.

The largest single Diesel-engine unit propelling any ship afloat develops not over 2000 hp. and there still remain many obstacles to be overcome before the single-acting multicylinder, four-stroke- or two-stroke-cycle Diesel engine on board ship is supplanted by the double-acting engine. Vertical engines of the latter type are still in the experimental stage, and their future does not appear very promising on account of the serious mechanical complication of scavenging pumps and the multiplicity of scavenging valves in each cylinder head.

The lack of simplicity of these designs has prompted inventors and manufacturers to experiment with the so called "valveless" two-stroke-cycle type which presumably will become the large marine Diesel engine of the future. One valveless two-stroke-cycle experimental engine of 260 hp. is claimed to have shown an oil consumption of 0.375 lb. per i.hp.-hr., which at a mechanical efficiency of 60 per cent. would give 0.462 lb. per b.hp.-hr.; an economy as good as that of a regular two-stroke-cycle engine with scavenging valves. The mean effective pressure obtained reached 125 lb. per sq.in., which is remarkable in view of the extremely simple scavenging method employed.

Among the valveless, two-stroke-cycle engines, the Junkers oil engine has attracted much attention. The excellence of the form of combustion space is a great advantage in this engine, and gives it the distinction of being the best oil engine, at least from a thermodynamic point of view. Practical experience alone, however, will tell whether its unusual design and certain mechanical disadvantages will be offset by this advantageous feature. Only a few smaller engines, mostly for marine propulsion and of the twin- or three-cylinder vertical types have been built abroad. The tandem arrangement of cylinders which results in the same number of crank efforts realized in the ordinary single-cylinder, double-acting, two-stroke-cycle engines has so far been applied only on two 800-hp., three-cylinder vertical engines to be fitted in a vessel of one of the largest steamship companies. Aside from two experimental engines in Professor Junkers' laboratory, no horizontal engines of this type have been built and opinions abroad are much divided concerning the future of the stationary horizontal tandem Junkers engine.

The horizontal, double-acting, four-stroke-cycle Diesel engine in twin-tandem arrangement of 1600-2000 b.hp., which I had the privilege of examining at Halle, is a splendid piece of machinery and operates apparently very satisfactorily on tar oil with a small addition of ignition oil. Although it is claimed that such an engine can be sold in Germany for from $30 to $32.50 per hp. against $35 per hp. for a steam plant, it should be expected that the repair and maintenance cost will be much higher than even that of our present gas engines on account of the high pressures and temperatures and because of carbonization at partial loads which can be prevented only by scrupulous cleanliness.

The small- and medium-sized Diesel engines from 40 or 50 hp. in single-cylinder units up to about 600 hp. in four-cylinder units play an important part abroad. They have come to stay and have reached a high degree of perfection. It is amazing to note how many manufacturers of gas and steam engines abroad have taken up the manufacture of Diesel engines, because they found that the sale of suction gas-producer plants and smaller steam engines has fallen off alarmingly within the last few years.

The reason is plain: The single-acting, four-stroke-cycle, single- or multi-cylinder Diesel engine, but particularly the former, is comparatively simple in construction and operation, especially when its unequalled efficiency is considered. This engine is always ready to be put into service at a moment's notice and has no standby losses; it does not require upkeep and attendance of boilers or gas producers, and its cost compared with that of a steam or gas plant is reasonable. Moreover, it can be installed in the basements of buildings below occupied dwellings, boilers not being permitted in such places in Europe. One of the greatest advantages is that the actual fuel consumption of Diesel engines taken over long periods of operation does not materially exceed the guaranteed figures, whereas in gas producer and steam plants this excess is quite considerable.

While the Diesel engine was originally a vertical engine, the last two or three years have witnessed a remarkable development of the horizontal type. Experience shows that the fuel consumption is very little, if any, higher than that of vertical engines, but the horizontal type is preferred by the customer not only on account of its lower cost, but because of its greater simplicity and accessibility. Especially the horizontal Diesel engine

with Lietzenmayer's open fuel injection nozzle offers these advantages. The open nozzle is simpler, can be more easily taken care of by unskilled attendants, gives better results when burning heavy oils and is less sensitive than the closed injection nozzle of the original Diesel engine.

The only difficulty of which some users of small- and medium-sized Diesel engines sometimes still complain is caused by the discharge valves of the air compressors for fuel injection.

The price of oil fuel in any locality must be considered in connection with the prevailing coal price to determine whether a Diesel-engine installation is economical or wasteful. Cheap oil fuel will favor the Diesel engine only where coal is very expensive, as on the Pacific Coast, whereas high oil and low coal prices will exclude it from any part of the country where such conditions exist.

Considering interest on the investment, depreciation, cost of fuel, attendance, maintenance and repairs, cost of supplies, etc., it is evident that in Europe where fuel prices are from two to three times higher while labor is only one-half to one-third of that in this country, the item of fuel cost constitutes a large part of the total cost of power, particularly since the other items, such as interest and depreciation, attendance and repairs are actually much lower and thus form a still smaller percentage of the total cost.

In this country conditions are practically reversed. Our fuel cost is not only lower and our labor cost much higher than in Europe, but the quality of our mechanical labor is not as excellent. Therefore, the fuel cost plays a much less important part in the total cost of power than the items into which labor largely enters, whereas the lower standard of manufacture and especially of maintenance favors the use of simple, although less economical, machinery.

The formation of syndicates and other influences in Europe have caused the price of oil to go up nearly 100 per cent. within the past few years, and, although this has somewhat retarded the steady growth of the Diesel-engine industry, it has by no means paralyzed it. An adequate liquid fuel has been found, however, in tar oil, a byproduct of the distillation of coal.

France now has to rely almost exclusively on the use of this fuel for Diesel engines, because of the high import duty on natural oils. In Germany a number of collieries have combined to form a company for the utilization of tar which operates a large factory at Meiderich, where, at present, over 300,000 tons of tar a year is distilled, producing over 80,000 tons of tar oil. This is contracted for in larger quantities at a price of approximately $12.50 per ton (5c. per gal.).

The heat value of tar oil is over 17,000 B.t.u. per lb. and its consumption in the Diesel engine is, therefore, not very much higher than that of natural oil varying now from 0.49 to 0.46 lb. per b.hp.-hr., according to the size of the engine.

The operation of medium-sized Diesel engines on tar oil, especially of those above 50 hp. per cylinder and with open fuel-injection nozzles, is satisfactory. For smaller engines the addition of a so called ignition oil is necessary.

Gas Turbines

Renewed activity is noticeable abroad in the matter of designing an adequate gas turbine, and I noticed that much interest is taken in this question by the most prominent manufacturing concerns in Europe. A large experimental gas turbine in Germany is operating splendidly, mechanically speaking, although the problem of satisfactory thermal efficiency still remains to be solved.

Gas Engines

The use of the so called "industrial waste gases" has progressed considerably. The four most prominent manufacturers in Germany alone are now building large gas engines for blast furnace and steel works at the rate of 120 per year and a number of twin-tandem engines of 5000 b.hp. operating very satisfactorily on rich coke-oven gas can now be seen in German coking plants.

The splendid results obtained with gas-blowing engines in Germany have prevented the turbo-blower from getting a firm foothold in that country, although its use in France, England and Russia is steadily increasing. Within the past two years the use of coke-oven gas for producing electric power by gas engines has greatly increased and after overcoming the many earlier difficulties the coke-oven gas engine is today just as safe and reliable as the blast-furnace gas engine. These difficulties arose principally from over-rating the engines, and from sulphur in the engine gas. Since the reasons for the trouble were recognized the causes, or their consequences, are now avoided.

The most prominent gas-engine builders are now governing their engines on a simple combination quantity-quality regulation principle. The valve-gear has been much simplified by all manufacturers, resulting not only in a reduction of the manufacturing cost, but also of operation; and gas-engine prices are, generally speaking, lower than they ever were before.

The gas cylinders, after a few unsatisfactory attempts to use cast steel, are invariably made of cast iron and with the exception of a few types are cast in one piece with an integral water jacket. The gas cylinders are often provided with hard cast-iron liners. The cracking of gas cylinders which formerly occurred too frequently is now much reduced by proper design, selected material and better pouring methods. Experience has taught that the life of a gas cylinder can be materially increased if the water jackets are thoroughly cleaned of all scale and muddy deposit at regular intervals of from four to eight weeks, depending upon the character of the cooling water used.

Gas cylinders on engines driving electric generators crack more frequently than those on blowing engines, because the former carry much higher loads and are usually shut down oftener. It was found that frequent shutting down and starting of gas engines shortened the life of the cylinders materially if the engines were allowed to cool off too much during shutdowns. The average life of a gas cylinder in the large steel plants in Germany is generally reckoned to be from five to six years.

The pistons are generally ribless cast-iron castings, although for large engines cast steel is used. One concern has recently introduced for blast-furnace gas engines one piece solid noncooled, forged steel pistons.

The piston rods are now usually of openhearth steel of about 100,000 lb. tensile strength, uncambered, and their diameter is approximately 26 per cent. of the diameter of the gas cylinders. Crossheads are made of forged nickel steel.

No uniformity of design can be claimed for piston-rod packings, but they invariably consist of single- or double-sectional cast-iron packing rings pressed against the rod by garter springs and separated by solid one-piece rings forming from 12 to 14 chambers. It is claimed that the clearances of the ring sections must be reduced to a minimum. So called fire rings are sometimes used, but several builders object to these solid rings.

The exhaust valves are of forged steel or Durana metal and are now almost exclusively of the noncooled mushroom type. Some builders, however, still prefer water-cooled exhaust valves for coke-oven gas engines.

Gas Cleaning Devices

The progress of the large gas engine in Europe has been accompanied by great improvements in the method of purifying blast-furnace gas. Until recently two systems of cleaning gas for engine purposes were largely used, the hydraulic fan and the Theissen washer. Results obtained with these appliances regarding both cleansing efficiency and cost of operation are satisfactory. Nevertheless, they now are superseded in Europe by new processes, which still further reduce not only the cost of the installation, but principally the cost of purification on ac-

count of the smaller power and water consumption.

One of these new methods is a radical departure from the former wet-cleaning processes and works on the principle of filtering the gas in a perfectly dry condition. The filtering is achieved by means of filter bags, made of special fabric and suspended vertically in several air-tight compartments. The raw blast-furnace gas coming from the furnaces at high temperature is first cooled by radiation to about dew point (around 130 deg. F.) and is then superheated 20 to 30 deg. F. by using any available source of heat, such as waste heat from gas engines, exhaust steam or even the sensible heat of the raw blast-furnace gas itself. By cooling the gas a large amount of moisture is removed and the scorching or burning of the filter bags is prevented, whereas the subsequent superheating avoids condensation of the remaining vapor and clogging of the bags.

The raw gas under the action of a fan passes from the inside to the outside of the filter bags, depositing all impurities on the surface. The layer of dust naturally increases continually and would eventually render the bags impenetrable, were it not for an automatic mechanism which shakes the bags at regular intervals of about four minutes and causes the dust to drop off. Simultaneously the compartment which is to be cleaned is automatically shut off from the others, and the direction of the gas current is reversed; the purified gas surrounding the filter bags now enters the latter, opening the pores of the filter material.

Gas Producers

It seems extraordinary that in spite of the high price of coal the development of the gas producer for power purposes has not kept pace with that of the gas engine in Europe. For many years the Mond gas producer with sulphate of ammonia recovery has been used extensively in England, and recently much interest was shown in Germany in plants that offer material financial advantages over the ordinary gas-producer installations. Since much sulphate of ammonia having a ready market as an artificial fertilizer, can be recovered from coal, the cost of gas making is considerably reduced, and installations are known where the saving thus obtained reaches $2 per ton of coal, which in many cases is as much as the value of the coal itself. The gas manufactured in such plants has a heat value of about 150 B.t.u. per cu.ft., and is of very uniform quality. On account of the removal of tar and dust, it is particularly adapted for gas engines, but its low cost makes its use profitable for metallurgical purposes.

The growing demand for coke has caused a remarkable increase in coke-oven plant activity. The days of the wasteful "bee-hive" coke oven are past, and the highly economical byproduct coke-oven is gradually taking its place. The manufacture of coke entails the accumulation of large quantities of coke breeze. Successful attempts were recently made to gasify this coke dust in special gas producers. One system operates with liquid slag very much on the order of a blast furnace, whereas another adapts a well known European producer design to the special requirements of gasification of very fine fuels.

Overloading Gas Engine

In one steel plant a novel system of overloading four-stroke-cycle gas engines 25 to 35 per cent. of their normal capacity is employed. This consists in scavenging ordinary four-stroke-cycle gas engines and introducing the fresh mixture under a pressure somewhat higher than atmospheric. This is achieved by raising the pressure of the combustion air and gas in turbo-blowers to a gage pressure of a few inches of mercury and by scavenging the spent gases of the previous power stroke with a blast of this air, subsequently introducing gas and air under slight pressure into the combustion chamber. By utilizing the waste heat of the gas engines for raising steam in suitable waste-heat boilers and driving steam turbines direct connected to the turbo-blowers, the power requirements of the auxiliaries can amply be supplied without resorting to outside sources of power, nor detracting from the increased output of the gas-engine.

Discussion

Following Mr. Freyn's report, W. R. Haynie, American representative of the Carels Frères Diesel engine, offered a few remarks substantiating in general the preceding speaker's remarks. Mr. Haynie expressed the opinion, however, that, although he had been unable to find any of the large Diesel engines talked about, he nevertheless believed that since commercial engines of 1000 hp. had been developed, it would be only a matter of time before 2000- and 3000-hp. units would be built successfully. He believed the future of the Diesel engine lay largely in marine practice. He also stated that many manufacturers considered 150 hp. per cylinder to be the commercial limit for four-stroke-cycle engines, although two-stroke-cycle engines could be built much larger.

Referring to oil fuels for Diesel engines, Mr. Haynie said:

"There is a question as to whether the heavier petroleums can be used successfully in Diesel engines. Generally, however, if the oil be free from noncombustible foreign matter, it can all be readily and profitably used.

"Among the Diesel-engine builders some raise the compression pressure and proportionately the injection pressure to create a higher temperature in the cylinders when using the heavier oils. This is not in conformance with the true Diesel principles. The purpose of creating heat by compression of air in the Diesel cylinder is to raise the temperature of the air to such a point that the process of combustion will take place.

"It may be stated that there is no petroleum residue of whatever heavy specific gravity, the ignition of which is not attainable by the temperature created by 30 to 35 atmospheres of compression. The combustion of the oil after ignition takes place is altogether another operation.

"The complete combustion of fuel oil in the cylinders after ignition by the temperature of the compression, is entirely dependent upon the process of handling the fuel before entering the cylinders. The period during which the oil has to be consumed is very short; in fact, a small fraction of a second; therefore, the process of combustion must take place almost instantaneously. The rapidity with which the oil is consumed, does not depend upon the specific gravity of the oil used, or its chemical properties, but upon the size of the particles or globules into which the oil is broken before meeting the air at ignition temperature.

"When using heavy residues in Diesel engines without reducing the size of the particles or globules, only the surface of the particles are consumed in the short period allowed for combustion, with the result that the interior portion is converted into a pitchy substance, which deposits on the surface of cylinders, piston and valves, causing first the use of a greater volume of oil than is necessary, and second the clogging of the cylinder and valves. Therefore, the oil should be completely atomized so as to effect complete combustion within the short period of approximately $\frac{1}{10}$ of the stroke."

Professor Magruder followed Mr. Haynie with some interesting statistics regarding the gasoline engine. These were to the effect that Great Britain has 150,000 motor cars for a population of fifty million, whereas in this country there are approximately 980,000 motor cars for a population of nearly one hundred million. This reduces to three and one-third cars per thousand people in Great Britain as against 10 cars per thousand people in this country; that is, this country has three times as many per thousand.

Further statistics given by Professor Magruder showed that the present annual output of gasoline in this country is sufficient only to operate 5 per cent. of the gasoline engines now sold and in operation in this country if run continually at their rated load.

In regard to the Diesel locomotive Professor Magruder stated that it was prohibitive as to price, and contained other detrimental features which made it thus far commercially impracticable.

Heating and Ventilation

Considered as power-plant problems. Layout and operation of systems and apparatus

Heating and Ventilation of Mount Vernon High School

By Ira N. Evans

The following is a description of the heating, ventilating and power equipment as designed for the new high-school building of Mount Vernon, N. Y. The building is now in the course of erection from plans of Architects Goldwin, Starrett & Van Vleck and the heating and power equipment involving several novel features in this class of work are under the supervision of Douglass Sprague and Associate, consulting engineers, of New York City.

Fig. 1 shows the building, which will have three stories and basement, and in plan will be about 300x100 ft., with an ell on the rear, 50x100 ft., for the gymnasium and auditorium. There are at present about 30 rooms with the usual space allotted for administration purposes. The toilet, locker and lunch rooms are in the basement. The south end of the building will be terraced and excavation made to form a sub-basement for the boiler, engine and fan rooms, with large coal bunker capacity. This removes all machinery from the basement proper, leaving it free for school purposes. The locker and toilet rooms at the rear of the building will be roofed over for the present, but at some future time these walls may be extended, allowing additional space for six classrooms on each floor or eighteen in all. A hot-water heating system under forced circulation is to be installed and capacity is provided at present to include the eighteen rooms without disturbing the present layout in any respect. Tees are to be left on the water mains for the additional risers and direct radiation and openings in the main air duct for the rising flues. Fig. 2 shows the piping system for one-half the building as it will be when these additional rooms are built, the other half being in duplicate.

Fig. 3 is a plan of the boiler and engine rooms. Two boilers of the return-tubular type, 60 in. in diameter and 17-ft. tubes, equipped with the Sidney Smith patent setting will be installed and will aggregate about 300 hp.

Each boiler will be provided with two circulating coils either side of the fire-box so arranged as to furnish prime heating surface, or provide means of heating the boiler-feed water before it enters the shell. This is especially advantageous when cold water is used.

A 6x4x6-in. feed pump will be provided for feeding the boilers through a closed feed-water heater and two filters with bypasses and valves. An injector is also provided. As all live steam will be returned to the boilers by gravity and all exhaust-steam condensation from the heaters can be utilized immediately, only a small receiving tank will be provided. There will be an automatic governor for the pump and a cold-water connection for makeup water. Due to the high temperature, 210 deg. F., and immediate return of condensation from the heaters, the steam requirements for feed water will be slight.

Each boiler will be provided with three nozzles: one for the safety valve, a 6-in. connection for the live-steam heater, and another 6-in. nozzle for the main steam header supplying the engines.

The high-pressure piping is made up of standard pipe and extra-heavy flanged cast-iron fittings capable of withstanding an operating pressure of 125 lb. if desired, although 80 to 100 lb. will probably be carried.

Bosses on the bottom of all tees in the main header will be tapped for a 1-in. drip connection to drain the main to a high-pressure trap in the engine room. This trap will also take the drip from the steam separators on the engines and discharge into the feed-water and exhaust heaters. There will be a line from each boiler for operating the feed pump independently in case the main steam header is out of commission. All steam connections from the boilers will have two valves, as shown, with open bleeders between the valves.

In the engine room two direct-connected units will be provided for lighting and power, one of 125 kw. and the other of 60 kw. capacity. A 10-in. pipe will handle the exhaust and this will communicate through an oil separator with the feed-water and exhaust-steam heaters. The piping will be arranged so that the steam may be bypassed around either heater to the atmosphere. The drip from either heater will be passed to the return tank shown underneath.

Domestic hot-water heaters situated back of the boilers will be arranged to use live steam through a reducing valve and will be trapped back to the feed-water heater as indicated in Fig. 3. A cast-iron tank heater will be provided for use with these hot-water tanks when the boilers are out of commission.

All steam and exhaust pipes, heaters, etc., will be covered with the best non-conducting material and in the sub-basement all return and supply water pipes of the heating system. The balance of the water piping will be exposed as heating surface and thereby reduce the temperature of the circulated water.

Fig. 1. Mount Vernon High School

Tables 2 and 3 show that one boiler will take care of the building in average winter weather above 35 deg. F. There will be less than 200 hr. when two boilers will be required. It should be borne in mind that when the greatest power is required during sessions, the sun is high and the outside temperature fairly warm. As school is in session only about 25 per cent. of the season requiring heat, there is plenty of time to make any necessary repairs between sessions, especial-

ly if the plant is in competent hands. This reduces the duplication usually necessary. With direct radiation and air supply for ventilation there will be practically two systems of heating, either of which could be utilized independently of the other if occasion required.

In addition to 40 cu.ft. per min. for each pupil, or a total of 70,000 cu.ft., air supply will be provided in the corridors, offices and administration rooms. In the corridors this is an innovation of the engineers who have found it advantageous in keeping the air of the building in better condition. The air supply will be taken through a window at the rear of the building which sets high and free from surrounding structures.

The hot-water indirect stack is composed of vento radiators two sections high and five deep, aggregating 6080 sq.ft. of surface. This stack will raise the air from 0 to 115 deg. with the water averaging 200 deg. The vertical flues to the different rooms will be set in the corridor walls, and the main air ducts will be run on the sub-basement ceiling in the boiler and engine rooms and under the floor of the basement corridor as the ducts pass to the north end of the building. A No. 15 multivane fan belted to a 40-hp. motor and of capacity to deliver 70,000 cu.ft. of air per min. against ¾ oz. pressure will be installed. The air velocity will be 500 ft. per min. in the flues and one-half that entering the rooms. Wire grills will be provided throughout for both air inlets and outlets. A separate duct under the main air duct will furnish the air supply to the gymnasium and auditorium. Dampers operated by air pressure from the automatic heat control system will be provided so that these rooms may be cut out when unoccupied.

Ventilating ducts from all of the rooms will lead to the attic space and into a trunk line duct connecting to each of two 84-in. propeller fans driven by 10-hp. motors. Provision will be made to discharge the air through a large ventilator on the roof. About six air changes per hour will be made in all classrooms. The provision of a power plant makes it possible to drive these fans by motors, thus gaining in economy over the use of small fan engines which are very wasteful of steam. Motors would also prove expensive if operated on public-service current at 6c. per kw.-hr.

A complete system of direct radiation composed of cast-iron radiators and 1¼-in. pipe coils operated on hot water will be provided to maintain the building at 65 to 70 deg. when school is not in session. The completed building will contain about 14,000 sq.ft. of radiation designed to furnish the requisite heat necessary in extreme weather with a maximum average water temperature of 200 deg.

In Fig. 2 is shown one of the special coils designed to eliminate the spring pieces and provide for expansion. This arrangement will allow space for the risers to run to the floor above without passing behind or in front of the coil. The system will comprise four main circuits: one for the north end with 4-in. supply and return mains, one for the south end with 3-in. mains, a 2½-in. supply and return for the gymnasium and auditorium and 5-in. pipes for the indirect stack. All returns will be brought back to the header near the pump with a separate valve and thermometer in each case. The supply mains will be run in the basement ceiling and the returns in the sub-basement. The north and south circuits will be divided into two 2½-in. loops, with valves and drawoffs connected at either end in all cases and so arranged that any one of the sections may be drawn down quickly in case of trouble without interfering with the remainder of the system.

The basement coils and radiators will be connected with 1¼-in. pipes in such a way that the drop-in head on the main 2½-in. line at the ceiling level will be sufficient to raise the water in the radiator when cold against the force of gravity and establish circulation.

All first-floor coils will have 1-in. connections shunted off the 2½-in. main, and 1¼x1-in. and 1½x1¼-in. risers will take care of the second and third floors in the same manner.

No valves will be provided on the risers as the two stop valves above referred to and drawoffs are considered sufficient in case of repairs. In all cases the riser connections will be crossed on the 2½-in. mains in such a manner as to enlarge the area by that of the riser when the surface is turned on.

The gymnasium and auditorium will be supplied by two circuits shown in Fig. 2 and so that the radiation on either side of the room may be drawn off or controlled separately. The radiators in the auditorium will have 1-in. connections shunted off the 2-in. main on the ceiling of the gymnasium. The 5-in. main to the indirect stack will be divided into five 3-in. connections so as to maintain as rapid a velocity through the stack as possible without too great a drop in head.

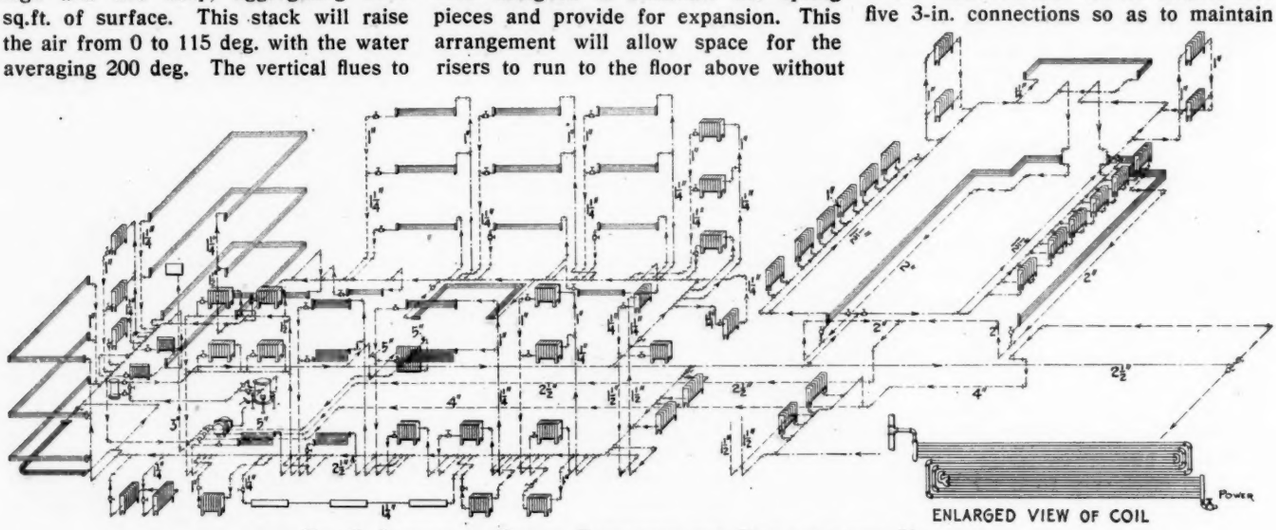


FIG. 2. LAYOUT OF DIRECT RADIATION FOR ONE-HALF THE BUILDING

In Figs. 2 and 3 are shown the connections to the heaters and pump on the hot-water system. The pump has a capacity of 1200 gal. per min. at 75-ft. head, and a 30-hp. motor and steam turbine are connected with a clutch so that the pump may be operated by either. The power may seem excessive but as all exhaust steam can be utilized, as per Tables 2 and 3, and all heat is returned in friction to the water, it is good practice in this case. The system is designed to give a maximum drop of less than 15 deg. under extreme conditions. The high velocities and capacity make it possible to use small mains. The reduction in size of the distributing mains in the north and south sections to 2½-in. minimizes the labor of cutting in tees.

The exhaust heater is 6 ft. 6 in. in diameter and 10 ft. long. The water is passed around the tubes and the steam through them on the counter-current principle. The live-steam heater is 3 ft. 6 in. in diameter and 12 ft. long, and is capable of taking care of the entire plant.

weather the thermostats will shut off the direct radiation. This will automatically reduce the power for the circulating pump on the water system to less than 25 hp.

A typical classroom will be taken as an example with 120 sq.ft. of coil surface, 1050 cu.ft. of air per min., a maximum radiation per square foot per hour

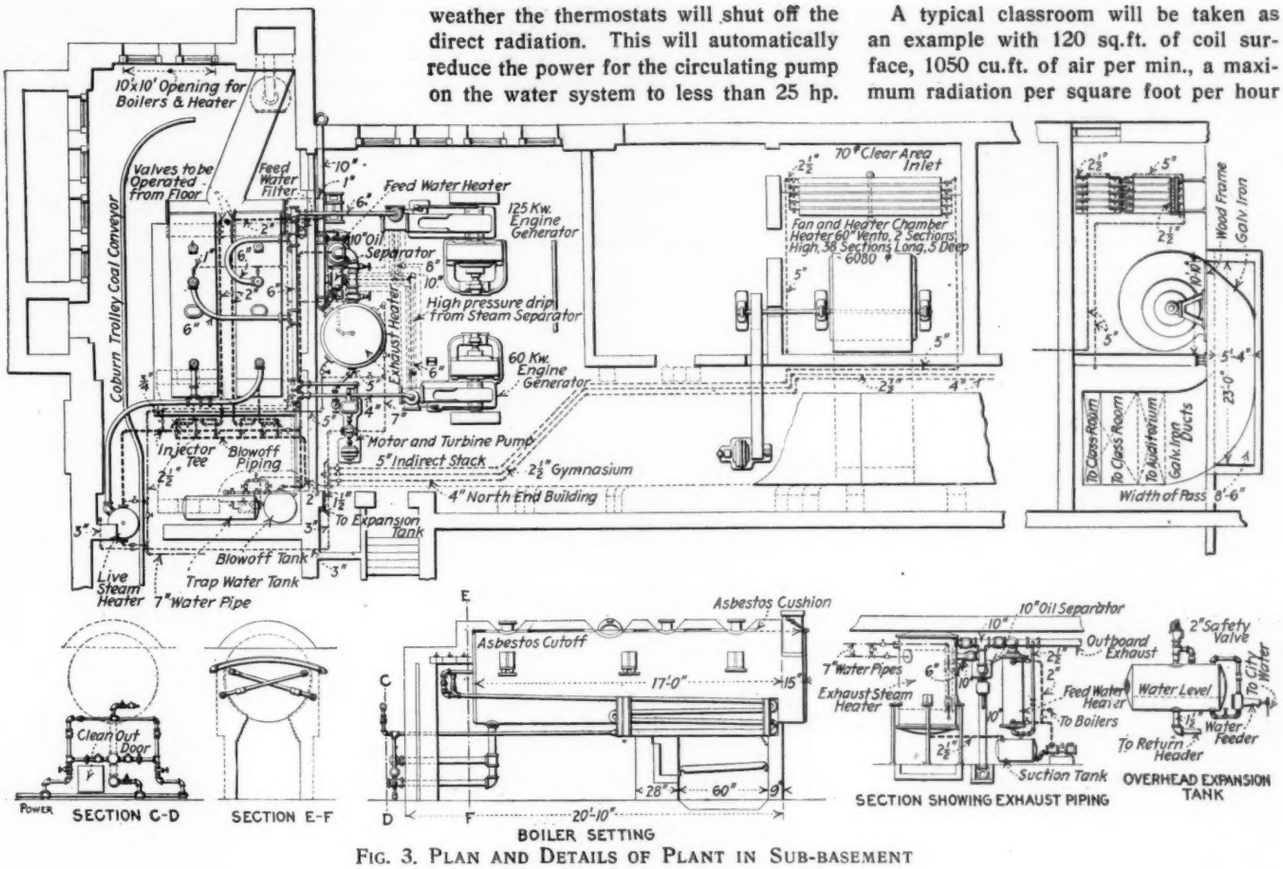


FIG. 3. PLAN AND DETAILS OF PLANT IN SUB-BASEMENT

It is provided with a gravity return to the boilers. A bleeder connection is made to the 2½-in. drip line so the condensation may be discharged to the feed-water heater and exhaust heater temporarily and then returned to the boilers by the feed pump. From the live-steam heater, the 7-in. water main will be run on the basement ceiling and reduced as the different circuits are taken off.

A complete automatic temperature-control system of the pneumatic type is provided for all main rooms on the direct radiation and to operate dampers. The indirect stack, however, is controlled by hand, as the residual heat in the water after the stack is shut off would interfere with automatic regulation. It is the intention to regulate the temperature of the entering air so that in all but extreme

Many of the rooms, such as toilets, will have direct radiation and no air supply. Consequently if the load on the direct radiation for the whole building were taken to find the necessary raise in air temperature to supply the heat radiated from the building when maintaining a temperature of 70 deg., it would bring the entering air up to 115 or 120 deg.

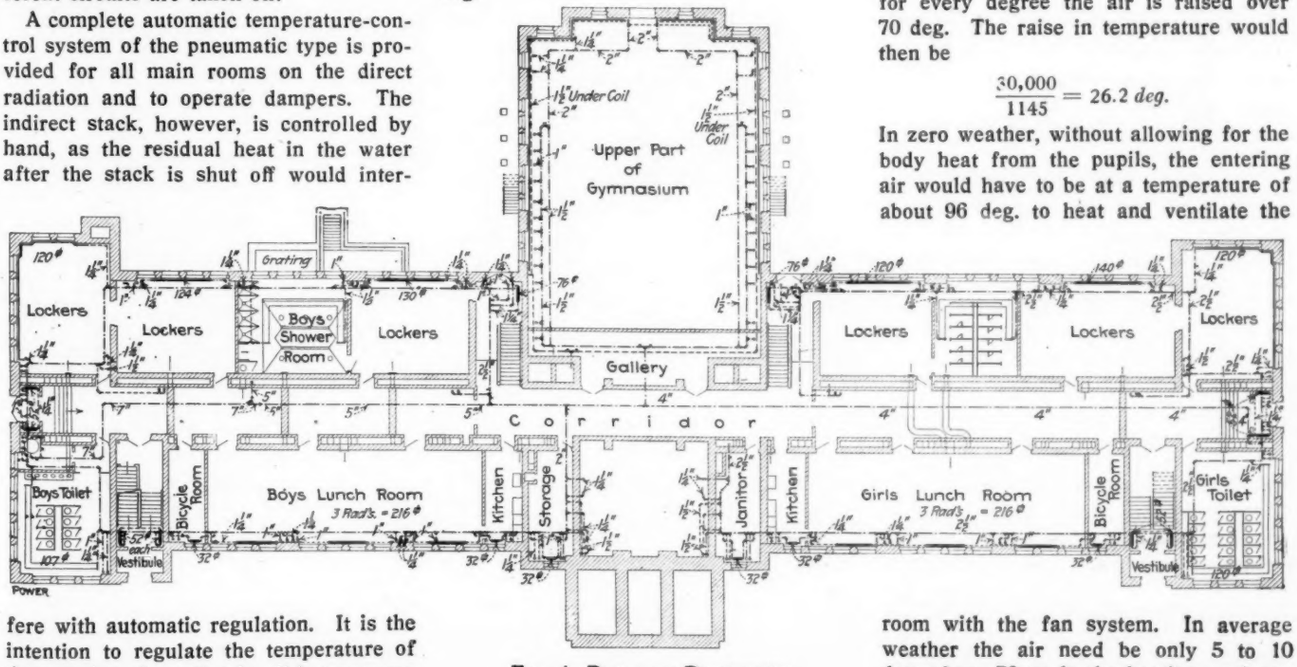


FIG. 4. PLAN OF BASEMENT

of 250 B.t.u., and 55 cu.ft. of air cooled or heated 1 deg. will require 1 B.t.u. Then.

$$120 \times 250 = 30{,}000 \text{ B.t.u. per hr.}$$

will be transmitted by the coils, and to supply the loss of heat per hour to the outside with the air will require

$$\frac{1050}{55} \times 60 = 1145 \text{ B.t.u.}$$

for every degree the air is raised over 70 deg. The raise in temperature would then be

$$\frac{30{,}000}{1145} = 26.2 \text{ deg.}$$

In zero weather, without allowing for the body heat from the pupils, the entering air would have to be at a temperature of about 96 deg. to heat and ventilate the room with the fan system. In average weather the air need be only 5 to 10 deg. above 70 to do the heating, and any

shortage may be made up by the thermostat turning on the direct radiation automatically.

Table 1, taken from an article in the Sept. 12, 1911, issue, shows the number of hours for each 10-deg. period of outside temperature during the heating season in the New York district. The data were obtained from the weather-bureau records. The entire season comprises 5034 hr. or 7 months. In school work there would also be two months; 40 days', operation after the heating season when only ventilation would be required and the temperature outside would be from 60 to 70 deg.

This school will be in operation from 8 a.m. to 1 p.m., 6 hr. per day, or 240 hr. operation after the heating season closes. During the seven months there will be in the present case 31 Saturdays of 5 hr. In the previous article, day operation was taken at 10 hr. instead of 6 and 5 hr. on Saturday. The school under consideration will then be occupied

$$(1545 - 155) \times 0.6 = 834 \textit{ hr.}$$

and 3192 hr. of night heating will be required. This does not include the 50- to 60-deg. period during nonsessions as presumably the plant will be shut down. Thus the school is actually in session about 20 to 25 per cent. of the heating season.

Tables 2 and 3 give the heat required for each outside period, the probable water temperature, transmission, boiler power and total steam required. The loss from the building is given for the different temperature periods and in Table 3 this loss is added to the heat for ventilation to obtain the total steam required. This is figured on continuous operation and, although the system may be shut down nights or over Sundays, the coal required will be the same and possibly somewhat more. The direct radiation is here figured operative during the 50 to 60-deg. period, but in practice it may be possible to shut down at that temperature, making some reduction in coal consumption.

The main engines will operate on 32 lb. of steam per i.hp.-hr. or on about 48 lb. per kw.-hr. at the switchboard. The total power required when ventilation is needed will be about 95 hp., including attic fans, supply fan and circulating pump. At 32 lb. per hp.-hr. this would provide 3040 lb. of exhaust steam per hour, which is just about what is required for heating in the 50 to 60-deg. period. During the 60 to 70-deg. period the exhaust will pass to the atmosphere as practically no heating is required.

At night the steam-turbine pump on the heating will require 20 hp., which at 60 lb. will require a steam consumption of 1200 lb. per hr. This is lower than any period in Table 2, so nothing is lost. In case street current were used and low-pressure steam with automatic heat control, the labor would be somewhat reduced, but possibly larger boilers or grates would be necessary to burn the coal at a lower rate. This coal would cost $6 per ton of 2000 lb., and 8 lb. evaporation will be assumed. The motor pump on the water system would be eliminated, leaving 60 hp. required by the fans for 1074 hr. Allowing 10 per cent. loss between boiler and engine raises the

TABLE I. HOURS FOR EACH TEMPERATURE PERIOD

Outside Temp. Deg. F.	Sessions		Nonsessions	
	Per Cent.	Hours	Per Cent.	Hours
0–10	0.60	5	1.60	67
10–20	3.25	27	6.13	257
20–30	13.25	110	17.83	750
30–40	32.40	270	27.82	1168
40–50	22.70	190	22.62	950
50–60	27.80	232	24.00	....
	100	834	100	3192
60–70		240		
Totals...		1074		3192

horsepower to 66, which practically equals 50 kw. Then the season load amounts to

$$1074 \times 50 = 53{,}700 \textit{ kw.-hr.}$$

Table 2 gives the total steam during nonsessions as 7,068,516 lb. and for sessions Table 3 gives 4,020,240, making a total of 11,088,756 lb. per season. Current at Mount Vernon costs 6c. per kw.-hr., so the charge per season would be

$$53{,}700 \times 0.06 = \$3222$$

With 8-lb. evaporation the season tonnage of coal would be

$$\frac{11{,}088{,}756}{8 \times 2000} = 693$$

which at $6 per ton would cost $4158, making a total of

$$\$3222 + 4158 = \$7380 \textit{ per season}$$

In Table 3 it is shown that the power will furnish excess steam only in the period of 60 to 70 deg. for the 240 hr. after the heating season is over and school is still in session.

In the case of the plant as installed, assume an evaporation of 7.45 lb. and pea coal at $4 per ton instead of an evaporation of 8 lb. and $6 per ton for egg and broken coal. For the fans 50 kw. will be required for 240 hr. At 48 lb. per kw.-hr. they would require

$$50 \times 48 \times 240 = 576{,}000 \textit{ lb. steam}$$

Subtracting the 91,440 lb. steam for the 60 to 70-deg. period, Table 3, gives 484,560 lb. to be added to the totals of Tables 2 and 3. The total steam would then be 11,088,756 + 484,560 = 11,573,316 *lb.* and the coal required

$$\frac{11{,}573{,}316}{7.45 \times 2000} = 776.7 \textit{ tons}$$

The cost of operating the entire plant would be

Coal 776.7 tons at $4	$3106.80
Extra water	10.00
Oil, waste and supplies	100.00
Labor of engineer and fireman	2000.00
Int. and Dep. on $6000 for engine, generator and piping at 10 per cent	600.00
	$5816.80

This shows a balance in favor of installing the engine and operating the entire plant of

$$\$7380 - 5816.80 = \$1563.20$$

It is not at all improbable that a large portion of this labor would be obligatory in either case. When nearly all the fuel and all the boilers are provided, operating an engine for power is highly economical. The above figures show the saving. If the difference in price of the two grades of coal had been $1 instead of $2, the difference in cost of operation on the same basis would be $526.60 less, or $1037.60. Even this is over 17 per cent. of cost of engine equipment.

It might be asked why coal at different prices is used. With pea coal the fires need attention constantly and when running the boilers for heating only, a fireman would be necessary. This would increase the labor charge. The egg or broken coal can be banked higher and left for longer periods.

TABLE 2. HEATING NONSESSIONS

Outside Temp. Deg. F.	Hours Each Period	Av. Temp Water Deg. F.	Temp. Water 65 Deg.	Trans. B.t.u. per Deg. Dif.	B.t.u. for 14,000 sq.ft.	Lb. Steam per hour	Boiler Hp. (34.5 lb.)	Total Steam Lb.
0–10	67	195	130	1.8	3,276,000	3276	95	219,492
10–20	257	185	120	1.8	3,024,000	3224	88	777,168
20–30	750	170	105	1.7	2,499,000	2499	72	1,874,250
30–40	1168	155	90	1.7	2,142,000	2142	62	2,501,856
40–50	950	140	75	1.7	1,785,000	1785	52	1,695,750
50–60	1008	130	65	1.6	1,456,000	1456	42	
								7,068,516

TABLE 3. VENTILATION AND HEATING (SESSIONS).

Outside Temp. Deg. F.	Temp. Air Ent. Rm. Deg. F.	Loss Ducts Deg. F.	Raise in Temp. Air Deg. F.	Lb. Steam (76.36 per Deg.)	Direct Rad. Steam for Heating, lb.	Total Lb. Steam per hr.	Hours Each Period	Total Lb. Steam Each Period	Boiler Hp. (34.5 lb.)
0–10	70	10	75	5727	3276	9003	5	45,015	261
10–20	70	10	65	4963	3024	7987	27	215,649	231
20–30	70	10	55	4200	2499	6699	110	736,890	194
30–40	70	8	43	3283	2142	5425	270	1,464,750	157
40–50	70	5	30	2291	1785	4076	190	774,440	118
50–60	70	5	20	1527	1456	2983	232	692,056	86
60–70	65	5	5	381	....	381	240	91,440	11
							Total.....	4,020,240	

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POWER

The Steam Specialty Man

Not infrequently, when the steam-specialty man sends his card to the engineer, or comes personally into the engine room, he is cut short before he has an opportunity to explain his proposition or show his sample and leave descriptive matter. He is regarded as a pest, unless the engineer is interested in his particular line or is in the market. There are times, of course, when the engineer is particularly busy and his mind is concentrated on the work in hand and no matter how interested he may be he cannot give any time to the salesman. As a rule, however, the engineer could devote a few moments to him, and he would find it a benefit to do so in the long run.

Many of the specialty men are practical men, who have been in the operating field. Such men are qualified through past experience to explain the operation of their specialty, practically and clearly and make useful suggestions. This practical explanation, with the descriptive matter given by the manufacturers of the article, serves to keep the engineer posted in recent developments in the business. It simply means additional knowledge to the engineer, which generally is superior to information gained from a textbook because it is explicitly explained by a practical man. Often he has a model or sample showing the construction and actual operation of a piece of new apparatus. An engineer may not have, or is unfamiliar with the article a salesman carries, but there is a possibility of his changing his position and finding just such a piece of equipment in the new plant—the information then comes in handy.

Besides this, the salesman is continually visiting plants and has the opportunity of seeing things and learning new kinks. He sees repair jobs being done, and the different ways of doing them. He notices some unusual wrinkle worked by an engineer on an emergency job and learns of new methods used in routine operation. As it is his business to interest the engineer, naturally he usually talks shop. He will mention he saw such and such a scheme carried out today and explains how it was worked. In this way the engineer obtains some more information he is not apt to find in books, and which may prove valuable at some future time.

On the whole, devoting a few moments to the specialty man is a decided benefit. He will not go away disgruntled because he did not make a sale at the time, for he left the engineer interested and gained a prospect who will remember him when later in the market, or who will present his proposition, if a good one, to the employer when the right time arrives.

The least the engineer can do is to be civil to the specialty man, and if too busy to see him at the time, invite him to call again. It pays in the long run. One way in which the specialty man sometimes proves of direct personal service to the engineer is in acquainting him with a new or better position, being very likely to hear of such opportunities.

Give Firemen a Chance

There is something about firing steam boilers that fascinates men who do that work just as there is about railroading or horses that fascinates those engaged in occupations where cars and horses are used.

Perhaps this is one reason why so many firemen have fired boilers all their working lives. The other and principal reason is, that once having learned to fire many men continue to do so because they could not work at other occupations and receive as much in wages as they do while firing. This latter condition is true of most men and plants.

To those not afraid of hard work and who simply want a job, the boiler room offers a good field. But a progressive fireman should consider his work in the boiler room only as preparatory to qualifying him for entrance into the engine room of that plant or of some other plant. To wait for promotion to engi-

neer in the plant where he has learned to fire means too great a loss of time in most cases. If he is a married man he feels that he cannot afford to go into the engine room as oiler on a wage of two or three dollars less per week than he receives as a fireman. On the other hand, if he seeks a position as engineer in another plant he must either lie about his experience in the hope of being taken on or be promptly rejected if he tells the truth. The employer cannot be censured; he wants an engineer, not a fireman. But the fireman wants a position as an engineer, and the more competent, ambitious and self-confident he is the more he is likely to lie about his experience. And there is some excuse for doing so. He wants advancement and finds the channels of progress blocked to him, so he follows the next easiest course.

There is need of greater and better attention to this matter. Surely such conditions work great harm not only to chief engineers and firemen, but to the whole field of power-plant engineering. There should be a general understanding among chief engineers that they will see to it that firemen desirous of advancing in engineering are given the opportunity of acquiring the essential experience. This means that the fireman, or those who wish to advance, would have to do duty in the engine room or with the repair gang. It means, too, that the chief engineer must be sure that a fireman is sincere, so that no shirker would be allowed to do other work just because he wanted a diversion. It is, however, easy to detect a man whose purposes are false. If the helpers in the repair gang and also the oilers were allowed to alternate their work with that of the firemen occasionally, it would be of immense value to all, and incidentally it would help the engineer-in-charge because there would be men available who could successfully do more than one branch of work.

A plan of this kind would mean that a fireman would not need to lie to get a position he was capable of filling. The chief engineer would know where and how best he could use such a man before he hired him.

Best of all, perhaps, is the fact that a man naturally adapted to engineering, but whose economic circumstances prevent him from otherwise getting the necessary experience, would have open to him the means of quickly developing into a competent engineer.

We hope that engineers will work for the realization of the above suggestions. By doing so they will not only make good friends but good engineers.

Falling Expansion Line with Superheated Steam

The author of a recent paper upon superheated steam is troubled by the fact that when superheated steam is used the expansion line of the indicator diagram falls away more rapidly than it would with saturated steam, and that, therefore, in order that the engine shall develop the same amount of power, the pressure must be raised or the cutoff be made later. This does not mean, however, that the engine is using more steam or more heat per unit of power developed; quite the contrary.

The holding up of the expansion line with saturated steam is due to the fact that when the steam is admitted to the cylinder, it gives up some of its heat to the containing surfaces, which have just been exposed to the temperature of the exhaust and are therefore cooler than the entering steam. Being only saturated, that is, having only enough heat to maintain it in the form of steam, it cannot give up any without a corresponding amount of condensation. At the point of cutoff there is therefore present in the cylinder, not dry steam but a mixture of steam and water.

If this mixture were expanded without taking up any heat from the cylinder walls or giving out any heat to them, the line would fall away considerably faster than does the line of the ordinary indicator diagram. As the expansion proceeds, some of the water which is above the boiling temperature at the lower pressure evaporates into steam, raising the pressure above what it would otherwise be. The expansion line of the diagram from an ordinary cylinder with saturated steam, does not represent therefore the relative pressure and volume of a constant amount of steam, but of a varying amount.

While this holding up of the expansion line adds to the area of the diagram and the power developed, it does so at an enormous expense. To heat up a cylinder with steam of boiler pressure and then to use a little of that heat at a lower temperature, to boil water to keep up the expansion line, allowing the greater part of it to keep on boiling out water after the exhaust valve is open, and to make steam to put into the exhaust, is a most inefficient way of fattening the diagram. The perfect engine would avoid this heat leak between the boiler and the condenser. A cylinder which absorbed no heat from and gave no heat to the working medium, would be the greatest improvement to which the steam engine is susceptible. The use of the steam jacket and highly finished internal surfaces are steps in this direction.

The use of superheated steam goes further than either of these. While it cannot preclude the absorption and giving out of heat by iron exposed alternately to high and low temperatures, this transfer is much less active between dry superheated steam than between moisture and the cylinder surfaces and with the use of superheated steam, the giving up of heat to the steam by the cylinder during the expansion is much less energetic and the expansion line is not raised so much above the normal as in the case of the saturated steam. But the heat which would be available for this purpose has not been taken from the entering steam in the first place, and if the author will compute the number of heat units which go to the condenser in both cases, he will find that, notwithstanding the longer cutoff or higher initial pressure, a smaller number of heat units are thus voided per horsepower-hour with superheated steam and its more rapidly falling expansion line than with saturated steam and the fuller diagram.

It speaks well for the carefulness of the boiler inspection in Montana that not a single boiler explosion has occurred during the present governor's administration (Gov. Norris' term of four years expires in January, 1913). In making his report, State Boiler Inspector J. H. Dailey showed also that the work has been growing increasingly difficult yet the department has not only been self-supporting, but in the past year has turned over to the state in excess of its expenses nearly seventy-five hundred dollars. Examples such as this one should be an inspiration to other states.

Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Oil Barrel Emptier Improved

Becoming interested in the device for lifting oil from a barrel as described by Mr. Ellsworth in the Sept. 17 issue, I tried to improve it as described below:

The materials used were a 1-in. tee, a ¾-in. short nipple and two 1-in. pipe bushings. One of the bushings was chucked in a lathe, and the wrench grip cut off, leaving the bushing about ⅜ in. long threaded the full length. This part of the bushing is shown at *A*, Fig. 1. Two holes *BB* were drilled in it so it could be screwed down into the thread on the tee. Then the lower end of the bushing *C* was faced off. The next step was to run the bushing *A* into the thread of the tee and screw the second bushing down to lock it in place. The tee was then "chucked" so that the holes in the bushing ran true. The size of this hole was then increased to allow a free fit for the brass tube *D*. The ¾-in. nipple *E* was then screwed into the other end of the tee and the ½-in. hose connection *F* was provided.

After the brass tube had been slipped through the tee the upper bushing *C* was unscrewed and some soft flax packing covered with graphite was wound down into the shallow stuffing-box. This joint proved to be a very good one. Fig. 1 shows the tee assembled.

This air oil lift is used in the same manner as that shown by Mr. Ellsworth. The tee and all its parts are attached to the barrel by screwing the nipple *E* into the bunghole of the barrel. Air under pressure is admitted at *F* and the oil flows from the top of the tube *D*. The outlet end of the tube *D* could readily be fitted with a hose connection if the oil had to be conveyed to a distance which. of course, is limited by the air pressure available.

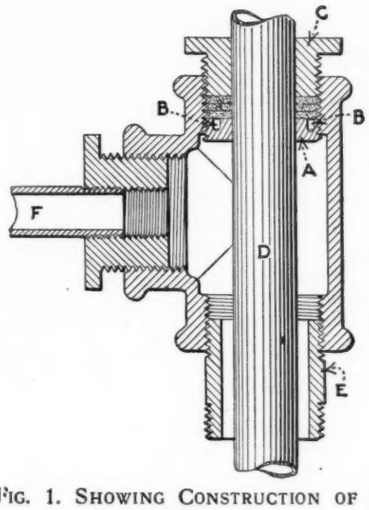


FIG. 1. SHOWING CONSTRUCTION OF AIR OIL LIFT PUMP

When one has to open an oil barrel to use a device of this kind, a bunghole auger, Fig. 2, is very handy. All the mill-supply dealers carry these tools, which enable one to easily open a barrel. The business end of this tool looks like an auger bit. Toward the top it assumes the form of a cone with a groove in it which forms the cutting edge. A few gage marks on this cone enable one to get a bunghole bored to fit any desired size of pipe.

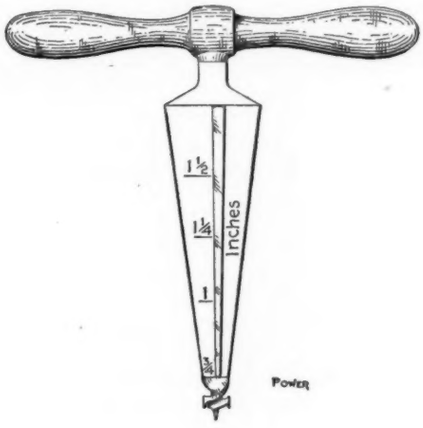


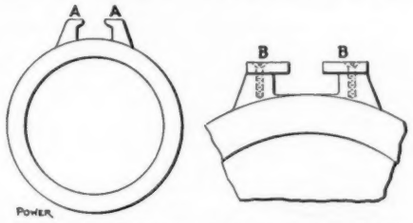
FIG. 2. BUNGHOLE AUGER

JOSEPH DEANE.

Cleveland, Ohio.

Repair to Worn Gate Valves

Recently we found that the lugs *AA* on the gates of some of our gate valves had become so worn that the nut on the end of the valve stem was likely to pull away from the gate, especially if the valve had been tightly closed.



SHOWING REPAIR TO GATE-VALVE DISK

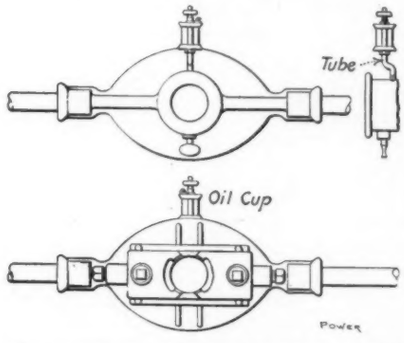
To avoid trouble we chipped and filed off the lugs. Two pieces of ¼-in. wrought-iron plate were then screwed on each side of the lugs, as shown at *BB*. The repair required but little time and the saving effected more than paid for the cost of making the repair.

G. B. JOSHI.

Poona, India.

Novel Oiler for Pipe Dies

Having quite a lot of pipe to thread, I thought of a plan that would enable me to dispense with the oil can that is such a nuisance when threading pipe. I hunted up a discarded oil cup and had the machinist drill and tap the stock (in the back) so that I could attach the oil cup. I bent a piece of copper tube and soldered it at the opening of the oil cup and also at the opening in the stock. The oil feed may be adjusted for different pipe sizes.



PIPE-THREAD CUTTING STOCKS FITTED WITH OIL CUP

Of course, there is room for improvement. The handles of the dies could be used to carry the oil and an adjustable feed could be easily obtained.

ROBERT GEILING.

Metuchen, N. J.

Oil Can Causes Turbine Over-Speed

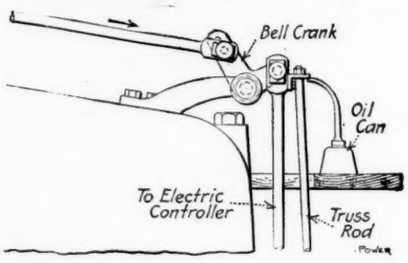
One Sunday afternoon, when the turbine was handling a load that often rose 40 per cent. above normal, one of the firemen suddenly became incapacitated, and a coalpasser was substituted. The engineer was obliged to visit the boiler room frequently to direct the new stoker. It was on one of these occasions, when the engineer had been delayed rather longer than usual, that a breathless

helper reported that the oiler was having trouble.

Hurrying to the engine room, the oiler reported he had tried to operate the emergency valve as the turbine was running away, but it would not move. An examination showed that the latch-lever had released the weight and only the fact that the packing of the valve stem had been tightened that morning had saved a shut down.

Just then the telephone bell rang, and the superintendent inquired if all were asleep and what had just happened to the voltage, as the recorder registered way above normal. The turbine then had dangerously overspeeded and the packing around the emergency valve stem was immediately slackened.

Ascending to the top platform the engineer reached for the oil can with which to oil the governor rod joints, but it was not there. Looking around, he found it laying on its side on the platform. Evidently the slight vibration of the turbine had caused it to fall from the bracket; but what made it roll so far to one side?



How the Oil Can Blocked the Governor

Then it was discovered that the oil can had fallen from the motor bracket, where it was usually kept, landing near the upright reach-rod; and the snout had caught under one arm of the governor-rod bell crank when the turbine was under a heavy load, as shown in the illustration, and had held the governor inoperative until the speed gave it sufficient power to spring the spout, and bounce the can to where it was found.

R. O. Richards.

Framingham, Mass.

Expensive Engine Repair

These indicator diagrams were taken from a 12x24-in. gridiron valve-gear engine, and the distortions are due entirely to the action of the governor, after it had been broken and improperly repaired.

This engine was installed in a woolen mill, and after running satisfactorily for over a year, the governor was accidentally broken. A machinist, called in to estimate on the repair, said it would cost about $20 to get the engine started again. The foreman (a skilled woolen worker) thought this sum too large so called in the mill blacksmith.

After "fixing" the governor, the engine refused to run. A bright thought struck the foreman. "Get one of them things that tells what's the matter with engines," said he. So a man and an indicator were sent for. After repeated trials the engine still worked badly. The entire load was then taken off the engine, when Fig. 1 was obtained.

FIG. 1

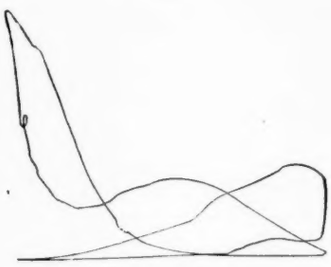


FIG. 2

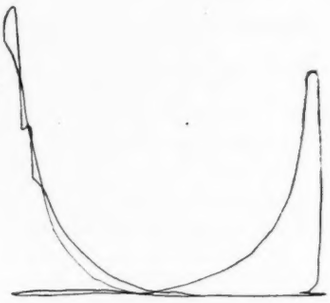


FIG. 3

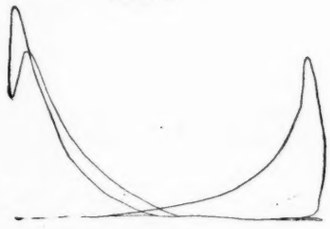


FIG. 4

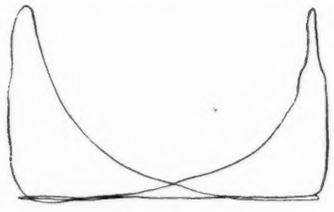


FIG. 5

Diagrams Taken during Adjustments to Valves and Governor

The indicator man then gave it up, and a machinist was sent for. After juggling with the governor and valves, the diagrams Figs. 2, 3 and 4 were obtained. The machinist then started an investigation, and located the trouble in the blacksmith's fixing, and later Fig. 5 was taken. The valves were then reset to their original position and good diagrams obtained.

The cost of the job was as follows: Blacksmith's time, 16 hr. at 32c., $5.12; helper's time, 16 hr. at 22c., $3.52; man and indicator, $10; machinist (shop charge), $20; total cost of the job, $38.64.

J. W. Thayer.

Newark, N. J.

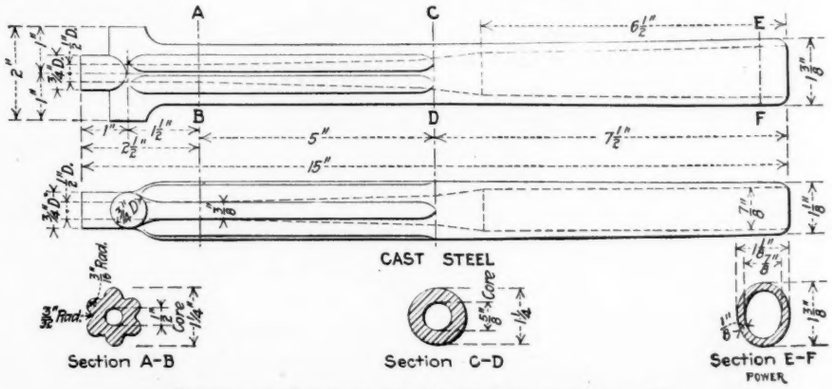
Strong Handle for Babbitt Hammers

The illustration shows a design for a durable handle for heavy babbitt hammers. When made of cast steel it gives long service, because of its strength. New hammers can be molded on the handles as required. The grip is cored out thin and shaped like the usual wooden handle for all kinds of hammers; therefore giving the same advantages in handling. The handle walls gradually grow thicker near the hammer and are ribbed, thus giving great strength where it is needed most. The first cost may be greater than for the same article in gas pipe or tubing, but the advantage to the workmen. and the long life of the handle more than make up for the additional cost.

When made according to the drawing the hammer will be of a good size for engine-room work, such as driving in keys, pins, etc.

W. W. Wysong.

La Porte, Ind.



Details of Handle for Heavy Babbitt Hammer

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Another Old Steam Plant

As a supplement to my article in the July 23 issue I would state that another very old steam plant still in operation has been brought to my notice, particulars of which may interest the readers.

This installation is at the Shipley Colliery Co.'s pits near Derby, England, and consists of a Cornish pumping engine of unknown origin. Authentic records show, however, that it has been in its present location and in satisfactory operation for over 100 years.

The principal dimensions and data are as follows:

Steam cylinder diameter (rebored 23 years ago)	52½ in
Stroke	7 ft.
Weight of beam	9 tons 1 cwt.
Two pump buckets, diameter	14½ in.
Water lift	195 ft.
Speed, strokes per minute	8 to 9 ft.
Duty	44,000 gal. per hr.

In addition to the duty of pit pumping, this engine also draws water from a pond half a mile away, delivering it into a reservoir for boiler feeding. A 10-in. ram pump is driven by the engine beam for this purpose.

The steam supply is the exhaust from a 12x24-in. and a 6x12-in. engine, which take steam at 70 lb. and exhaust against a back pressure of about 5 lb. into an old boiler which is used as a receiver. Steam for the ram pump is normally supplied by the exhaust from these two engines, but when this supply is insufficient, a makeup for the receiver is provided from the boilers through a reducing valve. A safety valve on the receiver is set to open at 8 lb. The vacuum is maintained by a jet condenser and averages about 25 in.

The maintenance costs for this engine are on a par with its steam consumption —low. The engine is said to use, per week:

1 gal. of cylinder oil.
1 qt. of machinery oil.
2 lb. of waste.

Bucket leathers last about three months and cost 14 shillings each. The labor cost for changing them is 15 shillings. Clack leathers last six months and cost 7 shillings each.

Such installations as this do indeed bear out the statement in the editorial of the July 23 issue, that the designers of these old engines "built better than they knew."

JOHN S. LEESE.

Manchester, Eng.

Burning Slack with Forced Draft

Reading V. L. Bowle's letter, "Replacing Stokers with Shaking Grates," prompts me to write of the change we made in our furnaces recently. We have been using the ordinary flat-surface herring-bone grates with natural draft for years. Our plant contains five horizontal tubular boilers, and we have been burning an average of 48 tons of Kentucky mine-run coal every 24 hr. Our evaporation by actual test showed 6.5 lb. of water per pound of coal; feed-water temperature 150 deg. The pressure carried is 100 lb. We recently installed a full equipment of slack burners using forced draft and now we evaporate 7.5 lb. of water per pound of slack. These burners are giving perfect satisfaction and we realize a saving of $25 per day.

J. W. DICKSON.

Memphis, Tenn.

Boiler Explosions

In the Oct. 8 issue, page 540, Lloyd V. Beets, in his letter on boiler explosions, suggests that boiler shells be drawn the same as tubes. This was discussed in POWER some nine or ten years ago by the late J. H. Dunbar, under the heading, "Comments on Power." At that time I believe the subject was called up by an explosion at Canton, Ohio. Several readers agreed that boiler shells would be better if lap welded the same as large size pipe. The trouble was that then no manufacturer made plates of the dimensions necessary to make a boiler shell of say 60 in. in diameter or larger, by the lap-weld process.

It is my opinion, borne out by considerable experience, that boilers made either lap welded or cold drawn, would be prohibitive in cost. If boiler shells were lap welded, they would not be as good as boilers made in the present way, as the metal would not be of a uniform thickness in all parts of the shell. Boilers of the horizontal return-tubular type have considerable stiffness added, due to the lapping of the joints. If made by the lap-weld or drawn process they would lack this feature, and would have a tendency to sag and become distorted when placed in the settings. Then there would be the difficulties encountered in trying to keep the shell round. In time, boiler shells, particularly the drums of water-tube boilers, will probably be welded by electricity or gas, similar to the welding of steam pipes as at present practiced in modern power-house piping, wherein screwed pipe fittings are eliminated on the larger sizes of pipes.

J. F. NAGLE.

Troy, N. Y.

Recollections of Corliss

Mr. Low's article on the Corliss engine in the Sept. 10 issue recalls anecdotes I heard about George Corliss when I was a boy living in Providence.

Mr. Corliss was known in his home town as a very shrewd business man as well as an inventive genius. He built the great Centennial engine and had it placed in the center of Machinery Hall, where it was seen and admired by everybody. Incidentally he made the exhibit a paying investment by taking contracts for furnishing power with it.

Most mechanical geniuses are poor business men, but Mr. Corliss always knew how to make a bargain, and had a very persuasive way of bringing people around to his way of thinking. I was an eye witness of the following incident:

In a large machine shop in Providence where machine tools are made, there was an engine built by another Providence builder. At the time the engine was put in, it was a single engine, but arranged so that another cylinder could be added if the works grew so as to need more power. Mr. Corliss called at this shop one day, to see about buying some tools for his shop. The proprietor took him all over the place, and into the engine room. At that time they were thinking of installing another cylinder, but first proposed to put on a wider wheel to take another belt. It was intended to have the original builder make the new wheel.

The proprietor told Mr. Corliss all about his plans, and they continued to talk until Mr. Corliss had brought the conversation around so there was hardly anything the man could say, but, "Well, Mr. Corliss, you can go ahead and make the new wheel," and incidentally, of course, he took a good order for tools. That engine is running today with a Corliss wheel, but all the other parts are of another make.

W. E. HOPKINS.

Torrington, Conn.

Improper Lubrication as the Cause of Air Receiver Explosions

Soapsuds for air-compressor lubrication have been much spoken of and written about, and Martin McGerry's soapsuds lubricator, as described in the Nov. 26 issue, being in the same issue with Letson Balliet's description of the explosion of an air receiver, on page 810, will make pertinent a few remarks on this subject.

Soapsuds are used for the very purpose of preventing just such an explosion as that described, but many operating engineers, and probably all air compressor builders do not recommend it. It is doubtful if there is much lubricating value in soap, and the effect of the alkali and the water in which the soap is dissolved is to rust the cylinder and its working parts. In Mr. McGerry's case, the use of oil for half an hour before shutting down may be the saving grace that not only prevents serious corrosion but furnishes enough real lubricant, even at long intervals, to prevent cutting.

No doubt, too, the water does much to prevent cutting, and it might be interesting to run the compressor with water only, leaving out the soap but still finishing off with oil, as before, to see how "slippery" such a combination would be. Many steam engines in the navy are often run without any cylinder oil.

These explosions of air receivers are altogether too frequent, and every operating engineer and owner should try to learn how to avoid them. Many who have studied the subject believe that the explosion is made possible by the presence of a mixture of oil and air, or perhaps finely divided carbon and air. Most, if not all, of these explosions occur in the receiver, and not in the compressor. This is peculiar when we realize that the air in the receiver must be cooler than at the instant of discharge from the air cylinder.

A single-stage air compressor, pumping to 100-lb. gage pressure delivers air at a temperature not much below 400 deg. F. In a small machine the relatively large jacketed wall surface cools this down almost instantly, so it is difficult to find such a temperature beyond the discharge valves. A compound compressor, at high-speed pumping to 100 lb. will deliver air in the pipe line at a temperature of not far from 275 to 300 deg. F.

These are not excessive temperatures for oil, except perhaps in the presence of the highly compressed oxygen in the air. Oils are made specially for air-compressor cylinders and only these should be used; yet many, either through ignorance or unwillingness to spend money on the proper material, are using entirely unsuitable oils.

Even with these special oils, however, explosions have occurred, probably due to an excess of air pressure and a very high temperature, or more probably to an excess of oil. Oil should be fed to an air cylinder as sparingly as will insure perfect running—and no more. An excess will produce a deposit of carbon on the valves, springs, ports, pipes and receivers, and may lead to a fatal explosion.

From the different actions observed, it would seem that these ignitions of deposited oil and carbon are of at least two kinds. Air receivers have been known suddenly to become hot to the glowing point, with the beginnings of "bagging" of the red-hot steel. Also, air pressures have been known suddenly to increase to double their original value, and then more gradually to return to normal, without a rupture, though the piping and receivers have been intensely hot. Then again, as in the case described, the ignition has been of the sudden or explosion type, with rupture and perhaps fatal results. In the first mentioned cases, it would seem that the burning had been slower, so that the increase of pressure was not so rapid but that the driven machinery could absorb it by increased speed, or the volume and area of the pipe and tank system were sufficiently large to relieve it fast enough to prevent rupture.

Stage compression is necessary for safety, when compressing air in large- or medium-sized cylinders to pressures above 100-lb. gage. The intercooling not only saves power, with proper design, but also reduces the final temperature and consequently the danger of ignition. With small cylinders, say 3x3 in., several hundred pounds have been reached in a single stage, but it is probable that little or no oil was used and the jacketing and radiation were sufficient to keep down the temperature of so small a quantity of air.

What sets off these explosions is hard to determine. Surely there cannot be any red-hot carbon on the springs, etc., because 400 deg. F. is not a red heat, to say nothing of the more usual figure of 275 deg. It must be started by a slow oxidation, perhaps like the spontaneous ignition in stored coal. If this were the case the explosion might be preceded by excessive temperature in the receiver.

This would seem to be borne out by the fact that the explosions nearly always take place in the receiver, where the air velocity has very materially decreased, rather than in the ports and pipe lines, where the higher velocity might not allow this gradual oxidation to take place. Unfortunately (or fortunately) no one is known to have observed this preëxplosion temperature.

At any rate we know these explosions do occur and probably from ignition of oil or its constituents, and so far the remedies, or rather the preventives, are:

Use only the right oil, made for the purpose, and use little of it; frequently remove all deposit from the springs, valves, etc., but do not use gasoline nor kerosene, or any other fuel oil with which to remove such deposits. Conditions will indicate how often this cleaning should be done; certainly once a month at the longest. Also drain the air receivers often.

H. B. SIDNEY.

New Orleans, La.

Test of Axial Flow Centrifugal Fan

In regard to the "Test of Axial Flow Centrifugal Fan," by Bratherick in the Oct. 22 issue, I was interested to note that he had discovered a fan that worked with 10 per cent. greater efficiency as a blower than it did as an exhauster. This came as a surprise and it seemed, from my experience in testing fans, that the mere reference of pressures and velocities to the fan proper would obviate any such difference in efficiency. A little computation brought this fact out and there may be readers who would be interested to follow these computations.

I noticed that the ratio of the air velocity to the rim velocity of the wheel bears a constant ratio of 46.6 per cent. This corresponds to a ratio of static pressure to rim-velocity pressure of 27 per cent., taken from a characteristic on a similar fan. This static pressure is due to the restricting of the discharge area of 71 per cent. approximately of the inlet-duct area (it is assumed that the area through the wheel casing is equal to the inlet-duct area). Knowing the rim-velocity pressure, the static pressure is obtained as per tabulation below:

Static Pressure	Air Velocity Pressure	Dynamic Pressure	Air Hp.	B. Hp.	Impact Mech. Efficiency
0.126	$\frac{1}{4}$	0.251	0.0286	0.057	0..0
0.172	$\frac{3}{16}$	0.369	0.0516	0.084	0.612
0.247	$\frac{1}{4}$	0.497	0.08	0.136	0.588
0.312	$\frac{1}{16}$	0.625	0.112	0.188	0.597

From the dynamic pressure and the volumes it will be seen that the air horsepower corresponding will be as given, which, with the input horsepower, gives the efficiencies shown in the table.

Therefore, these correspond almost exactly with the mechanical efficiency of the fan as a blower. This proves conclusively, at least to me, that the efficiency of a fan is no different when run as an exhauster than when run as a blower.

H. H. VALIQUET.

Hyde Park, Mass.

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it

Actual Diamensions of 8-in. Pipe

What are the actual inside diameter, cross-sectional area and cubical content of commercial 8-in. wrought-iron and steel pipe?

W. H. B.

Most American manufacturers quote the actual internal diameter of nominal 8-in. pipe as 7.981 in., making the internal cross-sectional area 50.027 sq.in., or 0.3474 sq.ft. One foot in length of nominal 8-in. pipe, therefore, contains 0.3474 cu.ft. or 2.5987 U. S. gal. (231 cu.in. per gal.), and 2.8785 lineal feet contain 1 cu.ft.

Rate of Combustion

A boiler evaporates 3211 lb. of water per hour. The grate is 6½ ft. by 4 ft. and 9½ lb. of water are evaporated per pound of coal. What is the rate of combustion?

R. M.

As 3211 lb. of water are evaporated per hour and 9½ lb. of water are evaporated per pound of coal the amount used would be

$$\frac{3211}{9.5} = 338 \text{ } lb. \text{ of coal per hour}$$

The grate being 6½x4 ft. would have an area of 26 sq.ft., therefore the rate of combustion would be $\frac{338}{26} = 13$ lb. of coal per hour per square foot of grate.

Safety Valve on Safety Valve

If the safety valve on a boiler is set to blow off at 100 lb., and then has its discharge connected into another safety valve set to blow off at 100 lb. with the discharge to the atmosphere, what pressure of the boiler will it take for the valves to blow off?

A. J. L.

Each safety valve being set to blow off for 100-lb. pressure per sq.in. more on one side than on the other side of the valve, then to lift the safety valve which discharges to the atmosphere, the pressure discharging from the valve next to the boiler must be 100 lb.; and as the pressure required to lift the valve next to the boiler must be 100 lb. more than the pressure on the discharge side of that valve, there would have to be 200 lb. pressure in the boiler to lift both safety valves.

To Remove Grease from a Boiler

How can deposits of grease be removed from a boiler?

R. R. D.

Grease can generally be cut and washed out by using soda ash, about $\frac{1}{10}$ lb. to a horsepower of the boiler for each application. With the boiler pressure down to zero, soda ash when pulverized can be introduced through the safety valve or other opening. Then with the boiler filled above the highest water level usually carried, the water should be boiled under a pressure of 5 to 10 lb. for about four hours, after which the boiler should be blown off and washed out with clean water. If the oil is coated or gummed to a considerable thickness, two or three such applications of soda ash may be necessary for thoroughly removing all traces of oil.

Danger of Ice Machine Racing

In a refrigerating plant using a Corliss engine type of ammonia compressor with belted governor, would there be danger of the compressor wrecking itself from excessive speed if the governor belt should break or run off at a time when the governor collar had been misplaced?

L. A. D.

If the governor became inoperative a dangerously high speed might be attained before sufficient compression could occur to check the speed. Excessive speed would occur in the event of a release of pressure due to a rupture from any cause, such as an accumulation of excessive compression of ammonia gas.

Degree of Superheating

What must be the degree of superheating of steam at 180-lb. gage pressure in order to show 50 deg. superheat after being throttled to 165-lb. gage pressure?

E. S. F.

There will be some loss of heat by radiation during throttling, but neglecting this loss, the total heat per pound of steam would be the same before and after throttling. Refering to Marks and Davis' steam tables, the total heat contained in one pound of steam raised from 32 deg. F. to 180 lb. absolute (which is practically the same as 165-lb. gage pressure), with 50-deg. superheat is 1227.2 B.t.u. and from inspection of the same tables it may be seen that for a pressure of 195 lb. absolute (which is practically equal to 180 lb. gage pressure), the same number of heat units would be contained by a pound of steam at the latter pressure for some degree of superheating between 40 and 50 deg. superheat, the number of B.t.u. for 40 deg. being 1223.3 and for 50 deg., 1229.2. As the number of B.t.u. for 40 deg. superheat is

$$(1227.2 - 1223.3) = 3.9 \text{ } B.t.u.$$

too small and as 1 deg. superheat at the pressure of 195 lb. absolute, 40 to 50 deg. superheat requires about

$$\frac{1229.2 - 1223.3}{10} = 0.59 \text{ } B.t.u.$$

per degree, then

$$\left(\frac{3.9}{0.59}\right) + 40 = 46.6$$

would be about the superheat required of the steam at 180-lb. gage pressure to be equal to 165-lb. gage pressure and 50 deg. superheat.

Horsepower for Feeding a Boiler

Neglecting losses from friction, what number of horsepower would be required for feeding 210 gal. of water per hour to a boiler carrying 86 lb. pressure per sq.in. and using a 4½x3x6-in. boiler-feed pump, the water being supplied to the pump at atmospheric pressure?

A. H.

The useful work would be the same using any size of pump. Assuming for present purposes that a pump is used having a water piston of 1 sq.in. area and making a stroke of 1 ft., then at each stroke there would be a displacement of 12 cu.in. of water, and pumping against 86 lb. boiler pressure each stroke being 1 ft. in length would require 86 ft.-lb. Therefore each cubic inch of water fed would require

$$\frac{86}{12} = 7.166 \text{ } ft.\text{-}lb.$$

Feeding 210 gal. of water per hour would be equivalent to $210 \times \frac{231}{60} = 808.5$ cu.in. per min., requiring $808.5 \times 7.166 = 5793.71$ ft.-lb. per min, or

$$\frac{5793.71}{33{,}000} = 0.175 \text{ } hp.$$

Engineers' Study Course

Simple Lessons and Lectures for Home Study

Involution and Evolution

These subjects are not so difficult to master as their names would seem to indicate.

Involution and evolution deal with the *powers* of numbers. A power of a number is the product obtained by multiplying that number by itself one or more times. For instance, 4 is the *second* power of 2, for

$$2 \times 2 = 4$$

27 is the *third* power of 3, for

$$3 \times 3 \times 3 = 27$$

256 is the *fourth power* of 4, for

$$4 \times 4 \times 4 \times 4 = 256$$

Thus we see that involution is the process of raising a quantity or number to any assigned power.

To let it be known to what power a number is to be raised we use an *exponent*. The *exponent* is the number that denotes how many times the number is to be taken as a factor, or to what power the number is to be raised. The exponent is written above and to the right of the number, as

$$4^2 = 4 \times 4 = 16$$

$$4^4 = 4 \times 4 \times 4 \times 4 = 256$$

The reader should notice that the exponent does not tell directly how many times the multiplication is to be performed (which is always one time less than the exponent) but how many times the factor itself must be taken. Thus 2^2 does not mean to multiply 2 by itself twice, but to multiply two 2's together.

From the expression 4^4 we see that by using the exponents we can express with two figures what would otherwise require four 4's and three multiplication signs. We also see that a product of equal factors is a power.

Powers of the same quantity may be multiplied by adding their exponents as,

$$4^2 \times 4^3 = 4^5$$

i.e., the product of 4^2 and 4^3 is 4^5, for

$$4^2 = 4 \times 4 = 16$$

and

$$4^3 = 4 \times 4 \times 4 = 64$$

Then

$$4^2 \times 4^3 = 16 \times 64 = 1024$$

which is the same as

$$4^5 = 4 \times 4 \times 4 \times 4 \times 4 = 1024$$

On the other hand, powers of the same quantity may be divided by subtracting the exponent of the divisor from the exponent of the dividend, as

$$\frac{4^5}{4^2} = 4^3$$

for $4^5 = 1024$ and $4^2 = 16$, and

$$\frac{1024}{16} = 64$$

which is the same as

$$4^3 = 4 \times 4 \times 4 = 64$$

Any quantity having the exponent 1 is equal only to itself; it shows that the factor is considered but once. But if the factor is already there we do not need an exponent because the exponent 1 does not change its value; hence, all quantities of the first power have no exponent.

The second power of a quantity is called a *square*.

It is so called because the second power of a number represents the *area of a square* of which that number represents one side. Take a square foot; each side is 12 in. long and we may imagine the area divided off into square inches, as shown in Fig. 1. Here each row represents 12 sq.in. and as there are

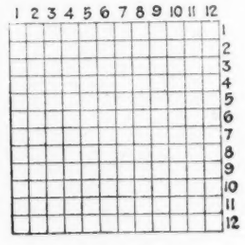


FIG. 1

12 rows the total number of square inches would be

$$12 \times 12 \text{ or } 12^2 = 144$$

For the same reason the third power of a number is called a *cube*, the volume of a cube being indicated by the third power of the number which indicates the length of its side. As an example, take a cubic foot, Fig. 2, the edges measure 12 in. each and each face contains 144 sq.in., as we have seen in our example in Fig. 1. If the cubic foot is divided into blocks of 1 cu.in. each, as in Fig. 2, it will be seen that the cube consists of 12 layers or slices and that there are 144 blocks or cubic inches to each layer or slice. Now the total number of blocks or cubic inches will be 12^3 or

$$12 \times 12 \times 12 = 1728$$

When a minus sign appears before or in front of an exponent it means that the number is to be raised to the power indicated by the exponent and written as the denominator of a fraction the numerator of which is 1. Thus

$$4^{-1} = \frac{1}{4^1} = \frac{1}{4}$$

$$4^{-2} = \frac{1}{4^2} = \frac{1}{16}$$

$$4^{-3} = \frac{1}{4^3} = \frac{1}{64}$$

To raise a common fraction to any power it is necessary to raise both of its terms or the numerator and denominator to the required power. Thus the cube of $\frac{2}{5}$ or

$$\left(\frac{2}{5}\right)^3 = \frac{2 \times 2 \times 2}{5 \times 5 \times 5} = \frac{8}{125}$$

Mixed numbers should be reduced to improper fractions and then both terms should be raised to the required power, the same as with proper fractions.

Example: What is the square of $24\frac{3}{4}$?

$$24\tfrac{3}{4} = \tfrac{99}{4} \text{ and } \frac{99 \times 99}{4 \times 4} = \tfrac{9801}{16} = 612\tfrac{9}{16}$$

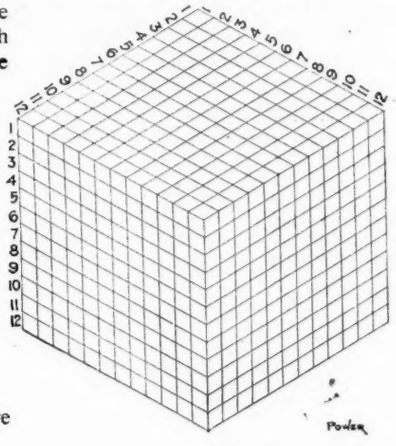


FIG. 2

The above $612\frac{9}{16}$ is the correct answer to $24\frac{3}{4}$ squared. Now suppose we had squared the whole number 24, and the fraction $\frac{3}{4}$ separately and then added the products, we would have obtained an incorrect answer for,

$$24^2, \text{ or } 24 \times 24 = 576$$

and $(\frac{3}{4})^2$ or

$$\tfrac{3}{4} \times \tfrac{3}{4} = \tfrac{9}{16}$$

then

$$576 + \tfrac{9}{16} = 576\tfrac{9}{16}$$

an incorrect answer.

This shows clearly that the sum of these two quantities squared is not the same as the square of the whole mixed quantity multiplied by itself.

The reason for this is quickly seen by squaring the number when expressed as a decimal, 24.75, for in multiplying this by itself it will be seen that the fractional part of the multiplier multiplies also the whole number part of the multiplicand and so increases the product over

what it would be if the mixed number were treated in the incorrect way as above.

Evolution

The factor repeated to produce a power is called a *root*, thus,

Four is the square of 2; then 2 is the square root of 4. Also 9 is the cube of 3; then 3 is the cube root of 9.

Evolution is the process of finding the root of a power, and is the reverse or opposite of involution, which is the process of finding a power from a root.

The radical sign $\sqrt{\ }$ placed before a number indicates that the root of that number is to be *extracted*. When the sign alone is used before a number it indicates that the *square root* is to be extracted. For other roots an index, which is a small figure, is placed *above* and to the left of the radical sign to indicate what root is to be extracted.

Thus

$\sqrt{9}$ indicates the *square root* of 9.

$\sqrt[3]{9}$ indicates the *cube root* of 9.

$\sqrt[4]{282}$ indicates the *fourth root* of 282.

The term under the sign, taken with the sign is known as a *radical* and may also be expressed by using a fractional exponent over the number as $4^{\frac{1}{2}}$, which is the same as $\sqrt{4}$. When a fractional exponent has a numerator other than 1, as $4^{\frac{3}{2}}$, it indicates that the number is to be raised to a power as well as having a root extracted. Thus in the example cited, $4^{\frac{3}{2}}$, the 4 is to be raised to its third power and the square root of the power is to be extracted.

$$4^3 = 64$$

and the square root of 64 or

$$\sqrt{64} = 8$$

so that $4^{\frac{3}{2}} = 8$. The operations may be done in either order and the answer will be the same for

$$\sqrt{4} = 2$$

and

$$2^3 = 8$$

Usually, however, it is simpler to raise to the power first.

Examples for Practice

(1) What is the square of 143; 16.84; $\frac{2}{3}$; $18\frac{3}{4}$?

(2) What is the cube of 14; 84.3; $\frac{5}{9}$; $24\frac{1}{8}$?

(3) (a) When the radical sign alone is used before a number, what is meant? (b) What does the index figure mean?

(4) What number is indicated by 24^2; 48^3; 18^4; 8^{-3}?

(5) What do you understand by $43^{\frac{3}{2}}$; $16^{\frac{1}{4}}$?

(6) How many square inches are there in a piece of sheet iron 18 in. square? How many square feet in a wall 14 ft. high and 22.5 ft. long.

(7) How many cubic inches in a cube that is 9 in. on an edge

Answers to Last Week's Problems

(1) $33\frac{1}{9}$.
(2) $17\frac{1}{2}$.
(3) $85.15.
(4) 6 in.
(5) 11.42 hr.
(6) 165.6 lb.
(7) 6 in.
(8) 120 lb.
(9) 131,625 gal.

A Convenient Safety Valve Chart

By Amos J. Carr

The Massachusetts formula for the area of a safety valve reads:

$$A = \frac{W \times 70}{P} \times 11$$

where

A = Area of safety valve in square inches per square foot of grate surface;

W = Pounds of water evaporated per second per square foot of grate surface;

P = Absolute pressure at which steam is generated.

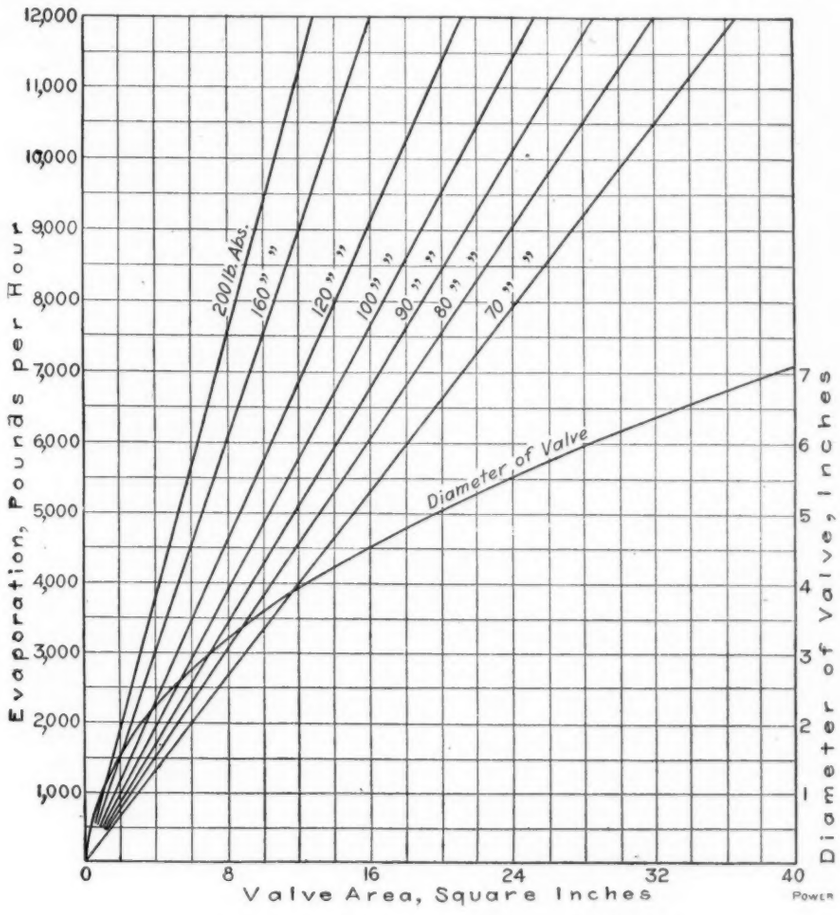
Anyone familiar with Napier's rule for the flow of steam through an orifice will readily recognize the quantity $\frac{W \times 70}{P}$ which is obtained by transposing, to solve for A, Napier's approximate formula

$$W = \frac{A \times P}{70}$$

Thus with the exception of the 11, the Massachusetts formula is identical.

It is known that $\frac{W \times 70}{P}$ will give the necessary opening to prevent an accumulation of pressure, but when it is considered that the effective opening of a safety valve is influenced by several factors and that the lift is directly proportional to the diameter or circumference, and does not bear any definite relation to the area of the valve, it is not quite clear how the factor 11 fits in. The following problem is solved on the assumption that the factor 11 has been so chosen as to give the desired results with a predetermined lift corresponding to a given diameter.

Take an evaporation of 5500 lb. of water per hr., or 1.527 lb. per sec. at an absolute pressure of 160 lb. per sq.in.



Safety-valve Chart for Different Pressures

Substituting the above values in Napier's rule and solving for A gives

$$A = \frac{1.527 \times 70}{160} = 0.668 \text{ sq.in.}$$

A 3-in. valve having a lift of 0.10 in.

gives a ring area of 0.94 sq.in., but with a seat of 45 deg. bevel the ring area must be multiplied by the cosine of 45 deg., or 0.71, which gives

$$0.94 \times 0.71 = 0.667 \textit{ sq.in.}$$

as the effective opening. This compares closely with the 0.668 sq.in. obtained by Napier's rule. The Massachusetts formula or the accompanying chart, assuming some grate area, will give the same result, so the factor 11 answers in this case.

A 2-in. valve having the same lift as the 3-in. valve in the above case would have a ring area of 0.63 sq.in. and an effective opening of

$$0.63 \times 0.71 = 0.45 \textit{ sq.in.}$$

From the chart it may be seen that this size of valve at 160 lb. steam pressure would be chosen for an evaporation of 2000 lb. of water per hour, or 0.555 lb. per sec. Napier's rule for a flow of 0.555 lb. of steam at an absolute pressure of 160 lb. per sq.in. gives an area of 0.24 sq.in., which is about one-half of the value computed above, assuming a 0.10-in. lift. A 2-in. valve must have a lift of 0.05 in. at the assumed pressure in order that the factor 11 shall suit this case.

Assuming the springs to have uniform compression in the two cases, the 0.05-in. lift with the 2-in. valve would be consistant, because the lift of the valve is proportional to the area multiplied by the pressure. The area of the 3-in. valve is 7.06 sq.in., and of the 2-in. valve a little less than half this figure.

Multiplying the 0.24 sq.in. obtained by Napier's rule by 11 gives an area of 2.64 sq.in., which would require a valve having a diameter of 1.75 in., the same as would have been obtained had the Massachusetts formula been used. A 2-in. valve would be required as there is no size between 1.5 and 2 in.

Construction and Use of Chart

As the evaporation of boilers is expressed in pounds of water per hour and W in the Massachusetts formula represents pounds of water evaporated per second per square foot of grate surface, the value of the factor W as used in an actual case would be.

$$\frac{W_{hr}}{3600 \times N}$$

where

W_{hr} = Pounds of water evaporated per hour;

N = Square feet of grate surface.

By letting A_t equal the total valve area, the formula becomes:

$$A_t = \frac{\frac{W_{hr}}{3600 \times N} \times 70 \times N}{P} \times 11$$

which may be simplified to

$$A_t = \frac{W_{hr}\ 0.21388}{P}$$

Assuming an evaporation of 10,000 lb. per hr., it is only necessary to move the decimal point of the constant 0.21388 four places to the right and divide by 160. In this case an area of 13.4 sq.in. would be required. From the bottom of the chart move upward on a vertical line corresponding to this area until the line opposite 10,000 is reached. Through this point and the zero point of evaporation and areas, draw a straight line and the area of the valve can be quickly determined for any evaporation at the above pressure.

By computing several diameters corresponding to chosen areas, points were established on the chart and through these the diameter curve was drawn. For example, a valve of 24 sq.in. area has a diameter of 5.53 in., which is the value given in the chart.

Should it be desired to know the size of valve for an evaporation beyond the limit of the chart it is only necessary to determine the area of the valve for one-half of the evaporation in question, double it and determine the diameter.

Take an evaporation of 24,000 lb. per hr. at 160 lb. abs. For one-half of this evaporation, 12,000 lb., a valve area of 16 sq.in. would be required; for 24,000 lb. the area would be 32 sq.in. An area of 32 sq.in. corresponds to a diameter over 6 in., so it will be necessary to go back to the 16-in. area, and a valve diameter of 4.5 in., of which two would be used. From the foregoing it is evident that the chart has a wide range of application.

Suppose a plant is being operated with a steam pressure of 120 lb. abs. and 10,000 lb. of water per hr. is being evaporated. An inspector reduces the working pressure to 100 lb. abs. The load demand being unchanged a higher evaporation would be required, this depending upon the change in engine speed, etc., to deliver the same power at a reduced mean effective pressure.

In the first case a 5-in. valve would be required and from the chart it can be seen that there is considerable margin, in that 11,000 lb. could be handled by the 5-in. valve at 160 lb. pressure, whereas assuming that 11,000 lb. must be evaporated at the reduced pressure, a 5.5-in. valve is necessary. Of course the inspector would decide this point, but the chief should be equal to the occasion and know the proper procedure.

State Fire Marshal Ahearn, of New York, reported recently that inspectors of his department had completed examination of every boiler in the state, to aid in strict enforcement of the law relating to the location of boilers in factories. He said the reports showed that a large majority of the boilers were safe. The work supplemented the efforts of the factory investigation commission.

Motor Driven Theater Fire Pump

The fire department of the city of New York in recognition of the necessity of adequate auxiliary fire-protective apparatus in theaters. requires the installation of a fire pump in each theater for use in connection with the regular city service in case of a fire.

A good example of the execution of this requirement is the installation at the Maxine Elliott Theater. The equipment consists of a Deming triplex 8x8-in. fire pump driven by a Westinghouse 20-hp. 220-volt direct-current motor.

The pump has a speed of 50 r.p.m., obtained through a double reduction gearing from the 1300 r.p.m. of the motor, and will deliver at this speed a 1¼-in. stream of water. The capacity of the pump is 250 gal. per min., at a pressure of 100 lb. As the plungers are bronze and the stuffing-boxes are brass lined, rusting cannot take place and the pump is therefore always ready for immediate operation.

The motor is controlled by an automatic starter which is connected through a pressure regulator to the hose reels so that when the hose is removed the pump will start automatically. This pump outfit came from the R. B. Carter Co., 152 Chambers St., New York City, agent for the pump.

For a National Museum

Interest has been revived in the plan to establish a national museum where the developments of engineering and the arts might be preserved. In the original design a large room in the Engineering Societies Building, New York City, was set apart for this purpose, but a lack of sufficient funds has prevented its accomplishment.

The recent visit of the German museum commission of Munich had much to do in interesting the members of the American Society of Mechanical Engineers in this project. The council of this society recently appointed a committee to coöperate with other organizations in establishing a great national museum. The committee is composed of E. D. Meier, H. G. Reist, Ambrose Swasey, George Mesta and George F. Kunz.

The Yarnell-Waring Co., Chestnut Hill, Philadelphia, has assumed management of the factory and sales of the Nelson Valve Co. The identities of both companies will be maintained and the management and officers of the Yarnall-Waring Co. will remain unchanged. D. Robert Yarnall will be vice-president and general manager and Bernard G. Waring, vice-president and manager of sales of the Nelson Valve Co.

Over the Spillway

Just Jests, Jabs, Joshes and Jumbles

A Merry Christmas to you, and yours! Charles Dickens said it all in these words: "I have always thought of Christmas . . . as a kind, forgiving, charitable, pleasant time; the only time I know of, in the long calendar of the year, when men and women seem by one consent to open their shut-up hearts freely, and to think of people below them as if they really were fellow passengers to the grave, and not another race of creatures bound on other journeys."

TIM HEALY'S JINED TH' SPUGS

Oh, Paddy, dear, an' did ye hear
Th' news that's goin' around?
Tim Healy's jined th' Spugs, bedad.
Sure, he's beat ye to th' ground!

Th' Spugs, says Tim, aare organized
To sthop poor girls from givin'
Christmas prisints to th' boss;
Thus raise th' cost av livin'.

An' whin Society tuk hold
To aid thim girls so maany,
'Twas Tim who gave th' thing éclat,
Wid Morgan's daughter Annie.

Oh, Timmy, man, av ye we're proud,
It's graand worruk, d'ye see?
Ye have th' roight idee, me bhoy.
Here's to ye, Timothy!

Man complains that there's a 60 per cent. discrepancy in the given values of skin-effect coefficients. This is a some serious complaint. Why not consult a physician?

It is proposed that "national and state commissions be formed to fix wages." Just now, when the wife of our bosom is striving to lift the mortgage off the old homestead, and Edythe and Maybelle are knitting our annual light-blue Christmas suspenders, and Cuthbert and Egbert are scrapping over who gives the pipe and who the Ivanhoe package—well, no fixation process will help us any; we're "fixed" already.

Describing a boiler explosion. a correspondent writes that "while the engineer was unhurt, he thought the Lord loved him, so he was through with the engineering field." Be of good cheer, brother, and hang on; nobody loves a quitter.

"Hey, you! drop that monkey-wrench and let the engine run if it wants to."—*Gas Review*. A most undignified editorial utterance, neighbor, but your sound common sense tickles us under the fifth rib. Go it! This is the one column where we don't have to be dignified, thanks be!

Power Plant Costs

By V. C. Bowles

In a power plant of almost any type and size, where modern equipment for measuring performance is not available, at least some sort of record covering operating conditions can and should be kept. Such records will eventually lead to the installing of suitable apparatus for obtaining complete and accurate data. There are very few owners, or managers, but will gladly coöperate with the engineers in such work.

The accompanying daily log and monthly chart show what can be done without the use of recording equipment.

The figures here given as examples are reasonably low, especially as the plant is anything but suited to the requirements of the service.

All data where any doubt may exist as to accuracy, such as volt and ammeter readings, air compressor revolution counter-readings, etc., are subject to a reduction of from 3 to 5 per cent. to eliminate all doubt.

The plant is that of the Clayton & Lambert Manufacturing Co., Detroit, Mich., and furnishes power, light and heat for manufacturing purposes, to eleven different buildings, covering about 8½ acres.

The power plant proper is in one of the main buildings on the outer edge of the group, and contains three vertical water-tube boilers of 100, 250 and 350 hp. respectively. The two larger boilers have stokers of the toboggan-grate type; the small one is hand fired. There are two boiler-feed pumps, one duplex 7x5x7-in. and one single 8x5x10-in., one 500-hp.open

POWER PLANT OF

CLAYTON & LAMBERT MFG. CO.

Date June 9th, 1911 **DAILY REPORT SHEET**

	DAY	NIGHT	SUNDAY & HOLIDAY
1. Average steam pressure.	100		
2. Average temperature feed water.	204		
3. Factor of evaporation.	1.045		
4. Cubic feet of water used by meter.			
5. Total lbs. water evaporated.			
6. Lbs. water evaporated from and at 212°.			
7. " " " " " " " per lb. coal fired.			
8. Average B. H. P. developed.			
9. Lbs. coal per B. H. P. H.			
10. Lbs. coal burned.	12000	2800	
11. " " " per square foot grate per hour.			
12. " ash removed.	780	220	
13. Per cent. ash.	6.5	7.8	
14. Hours run No. 1 engine.	.5	1.5	
15. " " " 2 "	10.5		
16. " " " 3 "	11	3	
17. K. W. H. delivered C. & L. switchboard.	1614	10	
18. " " Regal "	116	15	
19. Total K. W. H. generated.	1730	25	
20. Hours run air compressor.	11		
21. Cubic feet free air pumped.	55877		
22. Number hours exhaust steam on heating system.	11	3	
23. " " live " " " "			
24. Height of water in sprinkler tank.	92ft.		

REMARKS

CHEMICAL FIRE EXTINGUISHERS. See that they are in their proper stations and filled ready for action. Report at once any broken, unfilled or misplaced Extinguishers and their location.

VALVES. Post valves in yards and alleys.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
O	O	O	O	O	O	O	O	O	O	O	O	O	O	O	O
S	S	S	S	S	S	S	S	S	S	S	S	S	S	S	S

VALVES—Main Drip and Dry Line. If any valve is found shut or with strap broken, state reason for it and report what sprinklers it controls.

FIRE DOORS. Each fire door is to be examined and all cases of broken or weak latches, rails or hinges reported. Report any obstructions found against the doors tending to prevent their quick operation.

FIRE PAILS. Report name of room in which pails are missing from their places, or found partially empty, dirty, or not readily accessible.

WATER PRESSURE today at yard level 40 lbs. from Office Building

Is tank or reservoir full? Yes

Has water been shut off from any part of the sprinkler system since last inspection? No

If so, state reason for it and length of time it was shut off.

Report any cases of leaky sprinkler pipes or heads which appear to have been injured, or covered by corrosion.

GENERAL ORDER. Report here any oily waste, or accumulation of rubbish, or waste material found in any room.

Ernest Bein, Inspector. V. L. Bowles, Chief Engineer.

FIG. 1. FACSIMILIE OF DAILY REPORT SHEET

feed-water heater, one 5x6½x10-in. and one 5½x10x10-in. vacuum pump.

The generating equipment consists of one 125-kw. and one 250-kw. generator directly connected to single-valve engines, 16x15 in. and 22x20 in. respectively, and one 150-kw. machine directly connected to a 16x21-in. nonreleasing Corliss engine. There is also one 8 and 14 by 8 by 8-in. steam-driven air compressor.

The heating system contains about 60,000 sq.ft. of direct radiation on which a 17-in. vacuum is maintained. The demand for steam on the system from Oct. 15 to Apr. 15 is in excess of the exhaust steam available, and a pressure of 1½ lb. is required for various purposes, except nights and Sundays, the rest of the year. No charge is made, or credit given, for heat from Apr. 15 to Oct. 15, because no means are at hand for arriving at even an approximate value.

Kenward, W. Va., 1¼ nut, pea and slack coal at $2.30 per ton is used during the winter months. Repeated analyses have shown this coal to contain 2 per cent. moisture, 37 per cent. volatile, 57 per cent. fixed carbon, 3 per cent. ash and 1 per cent. sulphur and have a heat value of 14,400 B.t.u.

A saving of 7 per cent. was effected by purchasing this coal, as it proves better suited to the conditions during the winter, or heavy-load period. In the summer, or light-load period, a coking and slower burning coal gives better results.

No coal-handling apparatus is installed, due to the crowded condition of the plant; therefore all coal is handled by hand, and weighed on scales. On the coal sheet the steam pressure and feed-water temperature are also recorded hourly.

The amounts of water evaporation cannot be obtained, because of the absence of a meter, and the fact that the heating system is in operation the entire year, the returns being pumped direct to the heater. During the winter months the only raw water used is that which enters the condensers, as all steam, except that used for plating purposes is condensed and returned to the heater. The live steam used for plating is not taken into consideration in the accompanying figures, but is from 4 to 5 per cent. of the total evaporation.

Lacking watt meters, volt and ammeter readings are taken hourly to determine the electrical output. The load being comparatively steady, the results are reasonably accurate; 5 per cent., however, is deducted to be on the safe side.

The amount of air pumped is determined by using a revolution counter, and allowing 5 per cent. slippage. As near as the writer has been able to determine, 1000 cu.ft. of free air compressed to 90 lb. gage is equivalent to about 4 kw.-hr. and in reducing the work in pumping the air to an electrical equivalent, this figure is used. It is low, but is preferred to avoid exaggeration.

A draft gage is directly over the writer's desk in the engine room, so that by merely "plugging in" to a series of holes, the draft in either stack, or over any fire, can be seen at a glance, giving a very good idea of the condition of the fires at all times.

A daily report, or log sheet, showing the operating conditions of the plant is carried in duplicate, which is left on file in the engine room. On this sheet is also a record of conditions on the sprinkler system, and all fire risks. This is filled out by the man in charge of the heating system, as his rounds cover the entire establishment.

Due largely to the influence of wind direction and velocity, the coal and temperature lines do not correspond as closely as might be expected. Any deviation not accounted for by atmospheric conditions, therefore, must be due to changed operating conditions, and is the occasion for immediate investigation. A record of wind direction and velocity would add greatly to the value of these charts.

The value of the temperature record is illustrated by the following comparison.

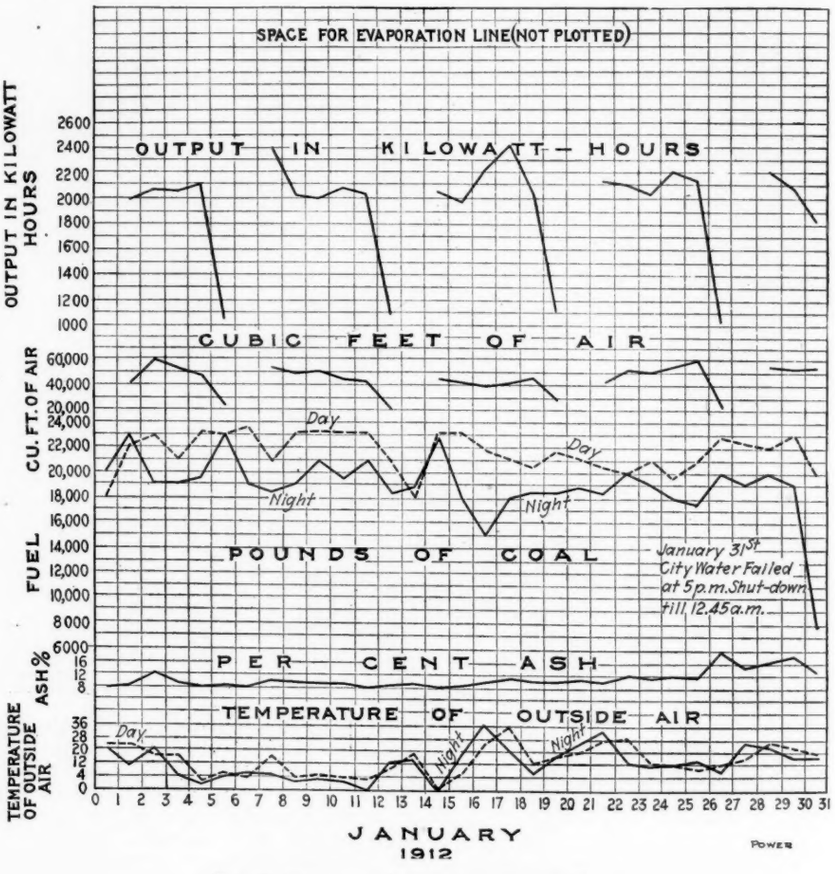


FIG. 2. MONTHLY REPORT IN CURVE FORM

A series of monthly charts covering every condition determinable by the means available, is placed in a convenient and conspicuous place in the engine room, for the employees and others interested to observe. The curves on these charts indicate the output in kilowatt-hours, cubic feet of air pumped, pounds of coal burned, per cent. of ash, and a line showing the temperature of the outside air for each 12-hr. period. Provision has been made for an evaporation curve to be plotted when a suitable meter is installed, and a like provision has also been made on the daily report sheets.

A study of the charts shows that the reducing valve admitting live steam to the heating system never closes from about Oct. 15 to Apr. 15, the coal curve following the temperature curve instead of the load curve.

During January, 1912, 20½ per cent more coal was burned than during the corresponding month of 1911, due principally to the lower temperature which prevailed in January, 1912. The mean temperature for January, 1911, was 27½ deg. and for January, 1912, 13½ deg., or 51 per cent. lower. The difference in coal consumption, however, was not due entirely to the lower temperature, for something like 500 ft. more radiation was added in the fall of 1911.

A complete set of books is carried by the writer, showing the amount and cost of supplies received, when and where used, and the amount, cost and every expense connected with the plant is kept track of so that monthly and yearly cost

sheets can be made up, showing the cost of operation and maintenance of every piece of apparatus in the plant.

Fixed Charge

The cost of the plant proper was $29,000, divided between power equipment, $17,000 and heating equipment, $12,000. For depreciation 5 per cent. and for interest and insurance 7 per cent. are allowed.

The heating equipment embraces all equipment necessary in the plant, if that for power had never been installed. Half of the fixed charge on the heating equipment is charged to power for six months of the year, because power derives benefit therefrom, although it is known that using exhaust steam from Apr. 15 to Oct. 15, more than offsets this figure. Hence it is charged against power, and heat is given the benefit of the doubt.

Half of the cost renewals to boilers, furnaces, stokers, piping and feed pumps, is charged to power, when in reality the greatest amount and cost of such renewals are necessary during, or after, the heavy-heating period, because then everything connected with the steam-generating end is driven to its greatest capacity.

Labor cost is divided so that in summer, or from Apr. 15 to Oct. 15, all service is charged to power, while for the remainder of the year, power is only charged with that necessary for the maintenance and supervision of power equipment, the rest being charged to heat.

As shown by the accompanying cost sheets, the load factor is very low. This, with other conditions which obtain, is hardly conducive to a low cost per kilowatt-hour.

The report sheets, charts and cost sheets, both separately and collectively, are working out very satisfactorily, and are developing new and valuable features every day. The accompanying copies are self-explanatory and will no doubt bring out points not touched upon here.

POWER PLANT COST RECORD

Clayton & Lambert Mfg. Co.

August 1911

	Power		Heat	
	Day	Night	Day	Night
Kw.-hr. at switchboard	39,570	2793		
Air reduced to kw.-hr.	3,556			
Total kw.-hr.	43,126			
Cu. ft. of free air pumped	1,482,547			
Coal used, tons	137.8	46.7		
Cost of coal used	$316.94	$107.41		
Cost of service	205.22	79.62		
Cost of oils, rags and supplies	24.11	12.06		
Cost of replacements to				
Boilers	0.85			
Stokers				
Furnaces				
Pumps	3.75			
Piping				
Engines	4.00			
Fixed charge	230.00		$60	
Total cost	784.87	139.09	60	
Cost of coal per kw-hr.	0.0073	0.0383		
Cost of service kw.-hr.	0.0047	0.0285		
Cost of supplies kw.-hr.	0.0002	0.0043		
Cost of replacements kw-hr.	0.0005			
Fixed charge kw.hr.	0.0053			
Net cost kw.-hr.	0.018	0.0498		
Net cost of all power per kw.-hr.	0.021			

January 1st, 1911 to January 1st, 1912

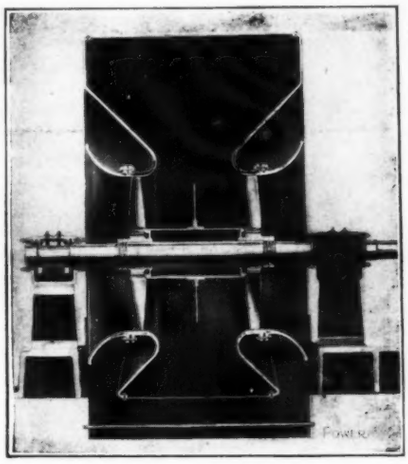
Kw.-hr at switchboard	475,641	50,390		
Air reduced to kw.-hr	57,795			
Total kw-hr	533,436			
Cu.ft. of free air pumped	14,448,990			
Coal used, tons	704.8	214.85	1392.4	1341.9
Cost of coal used	$1621.04	$494.15	$3202.52	$3086.37
Cost of service	1932.39	452.02	1095.56	988.04
Cost of oil, rags and supplies	247.29	127.16		
Cost of replacements to				
Boilers	1.75			
Stokers	154.93		197.49	
Furnaces	83.75		83.75	
Pumps	6.35		8.08	
Piping			9.30	
Engines	41.00			
Fixed charge	2400.00		1080.00	
Total cost	6488.50	1073.33	5676.70	4074.41
Cost of coal per kw-hr.	0.00348	0.0098		
Cost of service kw.-hr.	0.0036	0.0089		
Cost of supplies kw.-hr.	0.00046	0.0025		
Cost of replacements kw.-hr.	0.00054			
Fixed charge kw.-hr.	0.0045			
Net cost kw-hr.	0.01216	0.0213		
Net cost of all power per kw.-hr.	0.01295		All Heat	9751.11

Turbine Driven Blowers

These are built in two types, the smaller sizes with double helical runners and the larger with increase-pitch propellers. The manufacturers claim for both types a high unit economy of steam per air horsepower, especially on forced-draft pressures. Both types have the runners opposed and balanced for high speed and have a central deflector to prevent cross flow, as shown in the illustration.

The runners are removable endwise and may be reversed on the shaft for right- or left-hand discharge. The bearings are lead-bronze, or babbitt lined, ring-oiled and dust proof. The helical runner type has a solid cast-iron volute, and the propeller type is made with cast-iron sides and steel volutes.

The former is designed especially for forced draft for forges and furnaces and for use with underfeed stokers. The propeller type of blower has blades with an increase-pitch helical surface to reduce the shock losses due to impact at entrance. The kinetic efficiency of these blades is said to be high and in connection with the volute casing, reduces the discharge velocity to produce a pressure instead of eddies.



Section through Blower

Losses from back flow are prevented and a strong construction is had by dovetailing the blades into a solid hub. As the pull of centrifugal force is radial there is practically no blade distortion at high speed. A peculiarity of this make of blower is the use of interchangeable wheels in the same casing, to produce different pressures for approximately the same speeds and capacities. It is made by the McEwen Bros., Wellsville, N. Y.

Barometric Condenser Practice

By Miles Sampson

The barometer injector condenser has in recent years met with favor in those power plants which are so situated that a plentiful supply of cold condensing water is available, either by using pumps or a dam and its natural static head.

By using water under pressure for starting, this condenser will lift water from a depth of 20 to 25 ft. below the cones, condense the steam, and discharge into the hotwell against the atmospheric pressure. Forced injection, however, must be handled with great care. If the

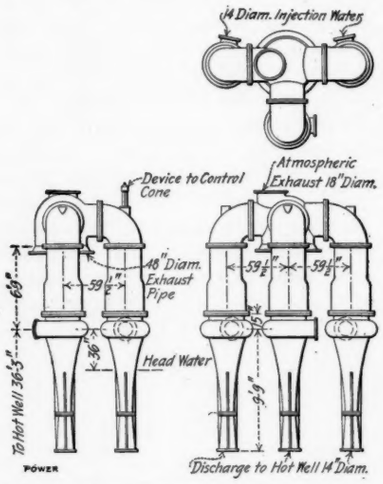
lift can be reduced to 10 ft. or less, the installation will be more satisfactory. By using a priming pipe between the injection and discharge pipes and by bypassing the cone, water is led into the discharge pipe, and a vacuum is formed. The water is then drawn over through the cones and the condenser is in operation. In such an installation the complications due to leaky tubes, power-using pumps and moving machinery are avoided, and the results obtained are surprisingly good, comparing favorably with the best pump equipments in less favored localities.

The triple-condenser arrangement as herewith shown serves a 3500-kw. horizontal turbine that is directly connected to an alternating-current generator. The turbine takes steam at 150 lb. gage and 50 deg. F. superheat. One winter day with a load of 4100 kw. and with the injection water at 36 deg. F., the vacuum was 28.4 as shown by a mercury column. At a steam consumption of 16.5 lb. per kilowatt-hour the total steam condensed would be about 68,000 lb. per hour. The total water passing per hour is probably not less than thirty times this amount, which is equivalent to a flow of 550 cu.ft. per minute. Such installations obviously require an ample supply of water.

It has been found necessary in practice to obtain certain conditions to insure a satisfactory installation. The levels are perhaps the most important feature, and should be carefully predetermined. The cone should be well above the head water, say, at least 3 ft. and at 36 ft. the hotwell overflow has been found to give the most satisfactory results. One company, to reduce the possibility of sucking water back into the engine, raises the exhaust head at least 4 ft. above the cone.

The hotwell should have a volume above the lower end of the discharge pipe equal at least to the combined volumes of the pipes above it. It will then be impossible for a slug of water to be drawn back into the exhaust pipe by the breaking of the water seal. If the hotwell is outside, provision should be made for draining it in cold weather.

Where possible, the injection piping to the cones should have no 90-deg. turns; these turns sometimes form air pockets and thus contribute to troublesome operation. The exhaust piping should be as short as conditions will allow. The double-deck arrangement, with the turbines placed over the boilers, has many advantages in the operation of this type of condenser.



TRIPLE BAROMETRIC CONDENSER FOR 30,000-KW. TURBINE

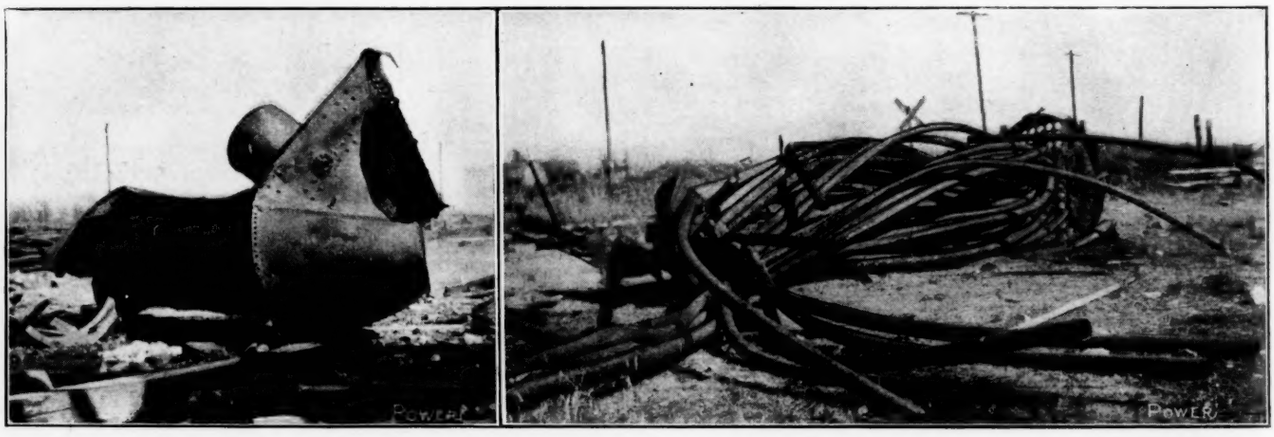
An interesting feature of the working of this arrangement is the unequal division of the load between the three legs of the condenser. With conditions seemingly identical, one of the three legs fails sometimes to carry hardly any of the load. In the instance cited, the temperatures of the discharge water were 60, 50 and 48 deg. F., and the divisions of the load were approximately 48, 28 and 24 per cent., respectively. One condenser was doing practically one-half the work. Careful repriming will sometimes help to equalize the load, but no positive remedy seems to have been found.

To recapitulate, the barometric condenser is simple and cheap to install and maintain; it is without moving parts and their unavoidable power consumption. It can easily be made to produce, in temperate climates, an average vacuum of 27.5 to 28 in. As this condenser requires only the observance of a few simple conditions and a plentiful supply of condensing water, it well deserves the recognition it has received from engineers.

Second Hand Boiler Explodes

A second-hand boiler exploded at Headric, Okla., at about 10 a.m., Nov. 8, seriously injuring two men and demolishing the boiler and engine. It was a 66-in. by 15-ft. return-tubular boiler and had been in service at Headric 11 years.

Apparently the explosion was due to corrosion and pitting of the shell longitudinally from 8 to 14 in. below the normal water line, where the shell was but ⅛ in. thick. The shell opened from one end to the other along this line. Both heads were torn loose from the shell, shearing some of the rivets and pulling some of them through the rivet holes in the rear head. The front head tore all the way around just outside of the tubes, leaving nearly all of the tubes connected to both heads. This portion of the boiler was carried about 100 ft.; the shell about 75 ft. in another direction, as illustrated herewith.



WHAT WAS LEFT OF THE EXPLODED BOILER

Nearly all of the braces pulled loose from the shell, or were broken.

The edges of the torn shell seemed to be crystallized and brittle. A working pressure of 90 lb. was carried, and the safety valve worked freely at 95 lb. The safety valve could not be found, and an examination of the shell showed that it had only been screwed into the boiler shell three threads.

The flywheel of the slide-valve engine was broken into several pieces, the hub and parts of the spokes being left on the shaft. The engine bed was cracked through just in front of the cylinder.

"Big Eats" for Chicago Engineers

The Hotel La Salle, in Chicago, one of the largest and best in the country, gave its entire staff of 1260 employees a banquet on Thanksgiving Day.

In order to avoid interruption of service the employees were fed in groups and at different hours. The engineering department was served at 1:30 p.m. and had a table set in the engine room because it would be impossible for the entire watch on duty to be absent from the plant at one time. Thus, instead of eating succulent turkey to the strains of an orchestra, the engineers ate it to the more familiar music of the industrious dynamos, engines and pumps.

Thirty covers were set, and in addition to the engineers, oilers, firemen and coal passers, the plumbers, machinists and electricians, who are also under the supervision of the chief engineer, sat down to the feast.

A generous menu was served, and, although there was a wealth of laughing and joking and good-natured banter, none neglected to do fullest justice to the good things provided. The occasion will be long and pleasantly remembered by all present.

Practical Refrigerating Engineers' Convention

On Dec. 5, 6 and 7 the Practical Refrigerating Engineers' Association held its third annual convention at Atlanta, Ga., with headquarters at the Kimball Hotel. The first session opened on Thursday afternoon. W. R. C. Smith, of the *Southern Engineer*, acting for Mayor Candler, welcomed the engineers to Atlanta and W. H. Ross, of *Ice and Refrigeration*, responded.

In his presidential address, F. H. Ladd, of Key West, Fla., spoke of the remarkable growth of the association. Starting in 1910 with but a scarce half dozen, no less than 67 regular members and a number of honorary and associate members had been enrolled at the 1911 meeting. During the past year the association had shown a healthy growth and it

PRACTICAL REFRIGERATING ENGINEERS AT ATLANTA

HOTEL LA SALLE ENGINEERS' THANKSGIVING FEAST IN THE ENGINE ROOM

is hoped that the meeting in Atlanta will add materially to the membership in the eastern section of the Southland. At present the membership is largely made up from Texas and Alabama. An examination is required for admittance and already owners and managers are showing their confidence in men belonging to the association.

Friday morning, W. H. Ross, in behalf of the American Association of Refrigeration, extended a cordial invitation to attend the third international congress of refrigeration to be held in Chicago during September, 1913. It was suggested that the Practical Association fall in line with others in the field and hold its annual convention in Chicago at this time. The opportunity of making the present organization a national association, as was originally intended, would never be better.

W. J. Rushton, who is an Atlanta man and president of the American Association, spoke briefly about the congress and the benefits to be derived by those attending.

While those at the convention realized the importance of attending in a body, September was a busy month for the refrigerating engineer and the trip was long. It was decided not to hold the annual convention at this time, but a committee was appointed to see that as many of the members as possible would go to Chicago.

With a view to extending the association at the congress, provision was made to amend the bylaws and provide for state chapters, the president or head of which would be a vice-president of the national association. The advantage of installing local chapters in such states as Illinois, Ohio and New York, where refrigerating engineers are more numerous, was fully appreciated and the association has great hopes of accomplishing its purpose at Chicago.

On Friday afternoon two papers on the steam boiler were read: one by Norman Register on its installation, care and management, and the other by William Francis on preparing a boiler for inspection. Next morning the question box, which had been left in the convention hall during the meeting, was opened and inquiries on soldering ice cans, measuring the contents of a tank lying on its side and calcium chloride losing its strength were fully discussed.

The entertainment program consisted of a visit to the Atlanta Brewing & Ice Co.'s plant where the conventioners did justice to an oyster roast with plenty of Atlanta's favorite brew on the side and incidentally a spirited contest between one of the negro employees and the brewery's goat. Keith's vaudeville was the attraction Friday night, and on Saturday afternoon the new raw-water ice plant of the Universal Ice Co. was inspected. In the evening an informal dinner at the Mechanics & Manufacturers Club brought the convention to a close.

Hot Springs, Ark., was chosen as the meeting place for 1913. The following officers were elected: President, Carl Nessler, Texas City, Tex.; vice-president, J. Higginbotham, Scranton, Miss.; secretary and treasurer, J. B. Embrey, Shreveport, La.; third place on board of directors, D. L. Gilbert, Dallas, Tex.; third place on board of examiners, W. A. Owen, Shreveport, La.

Car Heater Explosion

An accident of a serious nature, though fortunately not attended with disastrous results, occurred on a car of the Fort Wayne & Northern Indiana Traction Co. on the evening of Dec. 6. The car was passing through Logansport, Ind., about 11 o'clock and had just discharged the last passenger when the hard-coal heater of the hot-water heating system exploded with great violence, wrecking the forward end of the car and setting fire to it as well.

According to reports the motorman, who was protected by a partition, was thrown down amid a shower of glass, though not seriously injured. The heavy windows of the front end were blown out, the partition between the baggage and smoking compartments was torn loose and the heater was reduced to scrap. Had the accident occurred with a crowded car the results might have been very serious.

In answer to a request for an account of the accident and its causes, the company stated that it did not want to give out any information for publication, as it would not be fair to the manufacturers of the heater nor of any decided benefit to the users; users evidently meaning purchasers rather than the traveling public. About two years ago this traction system, then the Fort Wayne & Wabash Valley Traction Co., was forced to reorganize because of a succession of disastrous wrecks and consequent damage suits.

SOCIETY NOTES

The American Society of Agricultural Engineers has planned the program of its annual meeting at the Great Northern Hotel, Chicago, Friday morning, Dec. 27. Among the interesting topics will be "Isolated Gas Lighting Plants," by Eugene Becker, and "The Principles of Fuel Oil Engines," by C. F. Hirshfeld.

OBITUARY

GEORGE A. KIMBALL

George Albert Kimball, chief engineer of the Boston Elevated Railway Co., died on Dec. 3 at his home in Arlington, Mass., aged 62.

The transportation system of Boston was largely the work of Mr. Kimball. The Cambridge subway and elevated extension to Cambridge were designed and their construction was supervised by him. The layout of the street-car service in Boston and suburbs served by the elevated company is also largely the result of his efforts, as he was connected with the West End Street Ry. Co. before it was merged with the elevated.

In collaboration with Howard A. Carson, formerly chief engineer of the Transit Commission, and with Edmund S. Davis, the present chief engineer of that board, Mr. Kimball helped to design the East Boston tunnel, the Tremont street subway and the Washington street tunnel, which were built by the Transit Commission and subsequently leased to the elevated.

ALFRED P. BOLLER

Alfred Pancoast Boller, formerly chief engineer of the Manhattan Elevated R.R. and consulting engineer for the Department of Public Parks of New York, died on Dec. 9 at his home in East Orange, N. J. He was born in Philadelphia in 1840 and was graduated from the University of Pennsylvania in 1858, later receiving his degree of civil engineer from Rensselaer Polytechnic Institute, Troy, in 1861. He was associated in 1862 with the engineering work on the Lehigh Canal and later went with the Philadelphia & Erie R.R. In 1865 he became chief engineer of the Hudson River R.R.

Mr. Boller was vice-president of the American Society of Civil Engineers, president of the American Institute of Consulting Engineers, and a member of the Institute of Civil Engineers of London, England. He is survived by his wife, two daughters and three sons.

PERSONAL

Peter Neff, of Canton, Ohio, has resigned his position with the Arctic Ice Machine Co. to open an office as a consulting engineer. Mr. Neff will specialize in refrigeration.

J. C. McQuiston, advertising manager of the Westinghouse Electric & Manufacturing Co., on his recent retirement as president of the Pittsburgh Publicity Association was presented a handbag containing a silver-mounted traveling set in recognition of the work he has performed for the association.

S. Rosenzweig, mechanical engineer of the Erie City Iron Works, on his present lecture tour of Western universities, will include the universities of Cincinnati, Illinois, the Armour Institute of Technology and probably Purdue University. Mr. Rosenzweig will also speak before the A. S. M. E. branch of the Brooklyn Polytechnic Institute and Yale, Harvard and Lehigh universities on Jan. 4, 7, 10 and 14.